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## XVIII.

# GRAPHIC METHOD FOR DETERMINING THE INFLUENCE OF THE ANGULARITY OF THE CONNECTING ROD AND OF THE WEIGHT OF THE RECIPROCATING PARTS OF THE STEAM ENGINE.

By Passed Assistant Engineer A. B. Canaga, U. S. Navy.

The disturbing influence of the angularity of the connecting rod and of the inertia of the reciprocating parts can be located by kinematics, and measured by graphics much more readily and speedily than by the tedious and intricate processes of trigonometry. And in this article it is proposed to show how, by a simple use of the T-square, triangles, dividers, and scale, the proper rational relation between the pressure on the piston and the tangential pressure on the crank pin as modified by the inertia of the reciprocating parts, and by the angularity of the connecting rod, can be delineated clearly and satisfactorily.

In Fig. 1, let $O$ be the center of the crank shaft, $OB$ the crank and $AB$ the connecting rod.

The cross-head $A$ has a reciprocating motion along $OX$, and the crank pin $B$ a continuous uniform motion in the circumference $DBED'E'$.

Draw $AC$ perpendicular to $OX$, and produce the crank $OB$ un-

til it intersects $AC$. The point of intersection, $C$, gives the instantaneous axis of the connecting rod for its present position $AB$. Produce $AB$ to intersect the vertical diameter $E'E$ in $H$. The intercept $OH$ is to $OB$ as the velocity of the cross-head or piston is to the velocity of the crank pin; for, if

$V$ = velocity of the cross-head or piston, and

$v$ = velocity of the crank pin, then

$$AC : BC :: V : v. \qquad (1)$$

(See *A Manual of the Steam Engine and Other Prime Movers*, by Rankine, p. 511.)

In the triangles $ABC$ and $OBH$, the angle $ABC$ = the angle $OBH$, and the angle $BAC$ = the angle $BHO$, for $AC$ and $OE$ are parallel; the triangles are, therefore, equiangular and similar, consequently

$$AC : BC :: OH : OB, \qquad (2)$$

combining equations 1 and 2, gives,

$$OH : OB :: V : v. \qquad (3)$$

Let $P$ = total pressure on the piston, divided by the area of the piston in square inches.

Let $P'$ = total tangential pressure on the crank pin divided by the area of the piston in square inches.

The pressure on the piston multiplied by the distance $V$ it travels per second will equal the tangential pressure on the crank pin multiplied by the distance $v$ through which it travels per second; and as $P$ and $P'$ are to each other as the total pressures on the piston and crank pin, tangentially,

$$PV = P'v, \text{ or}$$

$$P' : P :: V : v. \qquad (4)$$

Combining equations 3 and 4, gives,

$$OH : OB :: P' : P. \qquad (5)$$

Now in Fig. 1, if $OB$ be drawn to scale to represent the *pressure* per square inch on the piston, $OH$ will represent to the same scale the tangential pressure on the crank pin due to the pressure $P$ per

square inch on the piston. That is, Fig. 1 drawn to a scale on which $OB$ represents $P$, would give the value of $P'$ in $OH$.

If, on the other hand, for equation 3, $OB$ be drawn to represent to scale the *velocity* of the crank pin in its orbit, $OH$ will represent to the same scale the velocity of the piston. That is, Fig. 1 is also a graphical expression of equation 3.

During the first part of a stroke of the piston the reciprocating parts are stored with an energy which is restored during the latter part of the stroke, and, in order to show graphically to what extent this transfer takes place, assume the proportions of an engine and the conditions under which it works, as follows:

Engine, horizontal, direct acting.

Diameter of piston, 20 ins.

Stroke of piston, 3 ft.

Length of connecting rod $= 4 \times$ crank $= 4 \times 1\frac{1}{2} = 6$ ft.

Revolutions per minute, 160.

Weight of reciprocating parts, 1,906 lbs.

Weight of reciprocating parts per sq. in. of piston $= 1006 \div \overline{10^2}\pi = 3.2$ lbs. $= W$.

Velocity of crank pin per second, $3 \times \pi \times 160 \div 60 = 25.13$ ft.

Scale of indicator, 1 in. $= 40$ lbs.

In this investigation the velocity of the crank pin is regarded as constant.

With $O$ as a center, in Fig. 2, and a radius $OD$ to some convenient scale—say, $\frac{1}{12}$ of an inch—to represent one foot of the velocity per second of the crank pin in its orbit, describe the circle $DED'E'$. Given in this scale $OD = \frac{25 \cdot 1.3}{12} = 2.09$ ins.

Divide the circumference of the crank pin circle into some even number of equal parts, say 20,* as this number gives 10 ordinates of the diagrams for each stroke. Draw $D'DX$ to represent the direction of motion of the piston. From $D$ and $D'$ lay off $Dd$ and

* The $\frac{1}{10}$ of an arc of a complete circle can be readily found by laying off from $D$ on the diameter $DD'$, Fig. 6, the distance $DL = \frac{1}{10} DD'$. With $D$ and $D'$ as centers and the diameter $DD'$ as a radius, describe the arcs intersecting at $X$. From $X$ draw the line $XL$, producing it to B, then the arc $DB$ will be approximately the $\frac{1}{10}$ of the circumference. The distance $DB$ can be used in the spacing dividers for stepping off the 20 divisions.

$D'd'$, respectively equal to four times the crank $OD$; $d$ and $d'$ will represent the ends of the stroke of the piston. From the points of division on the circumference 1, 2, 3, 4, etc., as centers and a radius $= 4 \times OD =$ length of connecting rod to scale, describe arcs cutting $OX$ in the points $1', 2', 3', 4'$, etc. These latter points will give the positions of the pistons when the crank pin is at the positions 1, 2, 3, 4, etc., respectively. The velocity of the crank pin being constant, the intervals of time required for the crank pin to pass these equal divisions of its orbit will be equal, and these small intervals of time will be to the second of time, as the small spaces in the divisions of the crank orbit are to the velocity of the crank pin per second. This proportion makes it possible to use the more ordinary and familiar equations of dynamics for expressing the relations of the quantities which are to be dealt with. When the crank pin is at $D$, the reciprocating parts have a horizontal velocity $= 0$, but when it arrives at the position 1, the reciprocating parts have a horizontal velocity, according to equation 3, represented to scale in Fig. 2 by $OH = V_1$.

The energy in foot pounds stored in the reciprocating parts per square inch of piston area is, therefore, $\frac{WV_1^2}{2g}$, and this energy has been imparted by a piston pressure $P_1$ acting through the distance $d1'$. Equating these quantities of work gives,

$$P_1 \times d1' = \frac{WV_1^2}{2g}. \qquad (6)$$

The mean pressure $P_1$ is that part of the piston pressure required to impart the velocity $V_1$ to the reciprocating parts while the crank pin moves from $D$ to 1, and while the piston itself travels from $d$ to $1'$.

Assume a mean tangential pressure $P_1'$ acting on the crank pin through the space $D1$ to impart the velocity to the reciprocating parts that is done by $P_1$ acting through $d1'$; then the work done by these two pressures is the same; that is,

$$P_1 \times d1' = P_1' \times D1. \qquad (7)$$

Equations (6) and (7) combined give

$$P_1' \times D1 = \frac{WV_1^2}{2g}. \qquad (8)$$

That is, a mean tangential pressure $P_1'$ acting from $D$ to 1, a distance equal to $\frac{1}{20}$ of a revolution of the crank pin $= \frac{1}{20} \times 3\pi = .47$ ft., causes the same energy to be stored in the reciprocating parts as a mean piston pressure $P_1$ acting from $d$ to $1'$.

The numerical values of the quantities in equation (8) are as follows:

$$W = 3.2 \text{ lbs.}$$

$$2 \times g = 2 \times 32 = 64$$

$$D1 = \tfrac{1}{20} \times 3\pi = .47 \text{ ft.}$$

$P_1'$ = mean tangential pressure on the crank pin between $D$ and 1 necessary to impart the velocity $V_1$ to the reciprocating parts. Making these substitutions, equation (8) becomes

$$P_1' \times .47 = \frac{3.2 \times V_1^2}{2 \times 32}, \text{ or}$$

$$9.4\, P_1' = V_1^2 = \overline{OH}^2. \qquad (9)$$

As already stated in drawing Fig. 2, $OD$ was taken so that $\frac{1}{12}$ of an inch represented one foot of the velocity of the crank pin. A solution of equation 9 would give $P_1'$ in twelfths of an inch, each of which would represent one pound pressure per square inch. But as the indicator scale on which 1 in. = 40 lbs. is to be used to measure pressures, the value of $P_1'$ is $\frac{40}{12} = 3\frac{1}{3}$ times too great. Now, if the constant part 9.4 in equation 9 be multiplied by $3\frac{1}{3}$, and the equality still holding, $P_1'$ would be reduced to the $\frac{1}{3\frac{1}{3}}$ part of itself, so that the ordinates to be subsequently found by graphics may be measured with the indicator scale directly. This does not alter the equality of the members of equation 9, for the numerical part is increased by $3\frac{1}{3}$ times itself, while $P_1'$ is measured by a scale on which $\frac{1}{40}$ of an inch, instead of $\frac{1}{12}$ of an inch, equals one pound. Equation 9 now becomes,

$$9.4 \times 3\tfrac{1}{3} \times P_1' = V_1^2 = \overline{OH}^2, \text{ or}$$

$$31.33\, P_1' = V_1^2 = \overline{OH}^2. \qquad (10)$$

In the change from equation 9 to 10 the left hand member of equation 9 has been multiplied by $3\frac{1}{3}$, and the same member will

be divided by $3\frac{1}{3}$ by using the scale on which 1 in. = 40 lbs. for measuring the value of $P_1'$ in the following graphical process, so that (10) is in reality a true equation.

Lay off $OJ$, in Fig. 2, equal to $\frac{31}{1}\cdot\frac{33}{2} = 2.61$ ins., $J$ will be the fixed point for the graphic solution of all subsequent equations for determining the influence of the inertia of the reciprocating parts. Take $OK = OH = V_1$, and through $k$ draw $kt$ parallel to $JH$, then $Ot = P_1'$, for by construction,

$$OJ : OH :: Ok : Ot, \text{ or}$$

$$Ot = \frac{OH \times Ok}{OJ}$$

$$OH = Ok = V_1, \text{ and } OJ = 31.33$$

$$\therefore\ Ot = \frac{V_1^2}{31.33} = P_1' \qquad (11)$$

When the crank pin is at $D$ the tangential pressure is zero, and increases in nearly a constant ratio until the crank pin arrives at the position 1, when the tangential pressure is such that the mean tangential pressure $P_1'$ will occur when the crank pin is near the point midway between the points $D$ and 1, and this approaches more nearly the truth as the number of divisions of the crank pin orbit are more numerous. Twenty divisions of this orbit give a near approximation for values of $P_1', P_2', P_3'$, etc. In order to find the true equivalent pressure on the piston of the pressure $P_1'$, bisect the arc $D1$, Fig. 2, in $a$. From $a$ with a radius equal to the length of the connecting rod to scale find the corresponding position $a'$ of the piston, and consequently the direction $a'a$ of the connecting rod. Through $t$ parallel to $a'a$ draw a line cutting $Oa$ in $m$. The distance $Om$ represents to the indicator scale the mean pressure per square inch on the piston required to accelerate the reciprocating parts while the crank pin travels from $D$ to 1. From $a'$ lay off $a'm' = Om$, above $OX$, the line of zero pressure. This pressure per square inch must be subtracted from the pressure per square inch on the piston, because its effort is wholly absorbed in imparting motion to the reciprocating parts, and has no immediate tendency to produce rotation of the crank. The equivalent of this pressure was, however, assumed to be act-

ing tangentially on the crank pin, because of the convenience of kinematic treatment. This energy imparted to the reciprocating parts is not lost, for after the crank pin passes the point of maximum velocity ratio of piston to crank pin, the energy of motion is given up, so that the inertia ordinates, found in a manner similar to $a'm' = Om$, must be added to the piston pressure ordinates in order to get the true effort which is transmitted to the crank pin.

When the crank pin arrives at position 2, the piston has acquired a velocity $V_2 = OH'$ to a scale of $\frac{1}{12}$ inch to the foot, same as before explained.

The energy in foot pounds now stored in the reciprocating parts $= \frac{WV_2^2}{2g}$, so that while the crank pin traveled from position 1 to 2, these parts have stored the total foot pounds stored at position 2 minus the foot pounds stored at position 1, or

$$\frac{WV_2^2}{2g} - \frac{WV_1^2}{2g}.$$

The foot pounds of energy stored in the reciprocating parts while the crank pin traveled from position 1 to 2, reduced to an equivalent energy at the crank pin, is $P_2' \times .47$. The numerical coefficient remains constant, while $P'$ varies for each division of the crank pin orbit. Putting these quantities of energy into an equation, and multiplying the left hand member—that is, the one containing $P_2'$—by $3\frac{1}{3}$, as was done before and for the same reason, gives,

$$9.4 \times 3\tfrac{1}{3} \times P_2' = (V_2^2 - V_1^2) = (V_2 + V_1)(V_2 - V_1),$$

or

$$31.33\, P_2' = (V_2 + V_1)(V_2 - V_1)$$
$$= (OH' + OH)(OH' - OH). \qquad (12)$$

This equation is solved graphically in a manner similar to that employed for equation 10, by laying off $Oh = OH' + OH$, and $Ok' = OH' - OH =$ the distance $HH'$; then through $k'$ draw a line parallel to $Jh$, cutting $EE'$ in $t'$. The intercept $Ot'$ on $EE'$ will give to the indicator scale the mean tangential pressure $P_2'$ on the crank pin between the positions 1 and 2 necessary to impart the increased energy which was stored in the reciprocating

parts. Bisect the arc 1–2 in $b$, and through $t'$ draw a line parallel to the connecting rod $b'b$ cutting $Ob$ in $n$, then $On$ will give the length for the inertia ordinate $b'n'$, which gives to scale the pressure per square inch on the piston required to impart the energy of motion to the reciprocating parts while the crank pin moves from position 1 to 2, and the piston moves from $1'$ to $2'$.

The foregoing steps indicate the process to be followed for each of the divisions of the crank pin orbit, from whence will result a series of ordinates of the inertia curves $m'n'o'q' - v'w'x'$ and $m''n''o''q'' - v''w''x''$, the former for the forward and the latter for the return stroke. The ordinates of these curves lying above $OX$, must be subtracted from, and the ones below added to the corresponding ordinates of the indicator diagram, giving new ordinates, which will represent the pressures per square inch on the piston, effective in producing a turning effort on the crank pin. The amount of this piston pressure which finally becomes at the crank pin tangential pressure, will be determined graphically later.

In the second and third quadrants of the crank pin circle, beginning at $D$ and going in the direction of the arrow, the connecting rod itself, instead of its prolongation, cuts $E'E$; this, for corresponding divisions to the right and left of $E'E$, gives shorter intercepts to the right in Fig. 2, and consequently the ordinates of the inertia curves are shorter at the inboard end of the diagrams. This does not affect the equality of the energy stored and restored during each stroke of the piston; for, although the ordinates are longer for the outboard end of the inertia diagram, the distance $dq'$ is less than $d'q'$, and the area $dq'm' =$ area $d'q'n'$.

The energy stored in the reciprocating parts as the crank pin travels through successive equal divisions, decreases from $D$, the beginning of the first quadrant, until the velocity ratio of piston to crank pin becomes a maximum, which for the ratio of crank to connecting rod of $\left\{\begin{matrix}1:6\\1:5\\1:4\end{matrix}\right\}$ occurs when the crank is $\left\{\begin{matrix}17\text{ min.}\\34\text{ min.}\\59\text{ min.}\end{matrix}\right\}$ towards $D$ from the position when the crank and connecting rod are at right angles, and this position for all practical ratios of crank to connecting rod is quite near enough; in fact, with the

ratio of 1 : 6, the difference is well within the limit of error of instruments and work.

The position at which the crank and connecting rod are at right angles can be found graphically thus: from $O$, in Fig. 3, lay off $OA$ and $OB$ at right angles, and equal to the crank and connecting rod, respectively, to some convenient scale.

Lay off $OC = AB$, and on $OC$ as a diameter, describe the semi-circle cutting the crank pin orbit in $D$; then $OD$ and $CD$ are at right angles, for the angle $CDO$ is inscribed in the semi-circle $CDO$. The point $C$, in Fig. 3, corresponds with the point $q'$, in Fig. 2, where the inertia curve cuts the zero line of pressure. This point is the same for the forward and the return strokes, neglecting, of course, friction of reciprocating parts, centrifugal action of connecting rod in its oscillations, and the vertical component of the weight of connecting rod.

The indicator diagrams shown in Fig. 4 may be regarded as assumed or they may really be the diagrams taken from the engine, the data for which was previously assumed, for the purpose of investigating the influence of the inertia of the reciprocating parts. The true pressure producing motion of the piston is the difference between the forward pressure on one side and the back pressure on the other side of the piston. That is, Fig. 4, $p1$ and $r2$, represent the unbalanced pressure per square inch on the piston when it is at the positions 1 and 2, respectively. At the right hand end of the indicator diagrams the back pressure for the inboard diagram runs above the forward pressure of the outboard diagram, giving an unbalanced back pressure represented by $u8$ and $v9$ for the positions of the piston coincident with these ordinates, respectively. From $dd'$, the zero of pressures in Fig. 5, lay off $p1''$, $r2''$, $u8''$ and $v9''$ equal to the ordinates having same letters in Fig 4. There results for the outboard end the true diagram $pruv$, the ordinates above the line $d'd$ indicating that the forward pressure is greater, and the ones below that the back pressure is the greater. A similar diagram is constructed (but not lettered) for the inboard end.

The ordinates of the curve $m'n'o' - v'w'x'$, Fig. 2, are subtracted from the corresponding ordinates of the true diagram $pruv$, Fig.

5, and the ordinates of the curve $x''w''v'' - o''n''m''$ of Fig. 2 are subtracted from the corresponding ordinates of the unlettered diagram for the inboard end, remembering that subtracting a negative ordinate is equivalent to adding a positive one. The subtractions in this operation are performed with the dividers. The inertia ordinates in Fig. 2 are subtracted from the upper ends of the corresponding ones of the true diagram in Fig. 5 from $d$ to $q'$, after which the ordinates in Fig. 2 becoming negative must be added to the corresponding ordinates of Fig. 5.

The ordinates of the resulting figures give the resultant pressure on the cross-head pin, which is effective in producing rotation of the crank.

The figure whose cross-section lines incline to the left is the diagram for the outboard, and the one with cross-section lines inclining to the right is the diagram for the inboard end. These cross-sectioned diagrams in Fig. 5 represent the indicator diagrams corrected for the inertia of the reciprocating parts. For convenience of reference these cross-sectioned diagrams will be termed *corrected diagrams*. The ordinates of these corrected diagrams are to the same scale as the original indicator diagrams; that is, 1 inch = 40 lbs.

A comparison of the corrected diagrams with the ones from which they were derived shows that an engine may have a statical stress at the instant of starting with full steam pressure, much greater than any stress to which it may be subjected when running.

The amount of energy transferred from the first to the latter part of each stroke by the reciprocating parts increases with the weight of these parts and with the square of their velocity. The influence of the weight of these parts would appear to suggest a means of equalizing the tangential effort on the crank pin by increasing the weight of the reciprocating parts as the number of expansions is increased. But employing increased weight of these parts for the purposes of equalizing the effort on the crank pin is bad practice for stationary and worse for marine engine design. The weight of the reciprocating parts will be sufficient when they are designed to withstand their proper working stresses. The initial pressure on the piston, the point of cut

off, and the amount of compression can be adjusted so that the tangential pressure will be near enough constant to insure smooth running of an engine. Compression, as shown by the horizontally cross-sectioned part of the corrected diagram in Fig. 5 is carried too far. There should be no part of the corrected diagram below the zero line *d'd*.

There yet remains to be shown graphically the influence of the angularity of the connecting rod on the effective piston pressure in its transmission to tangential pressure on the crank pin. When the crank is on its dead center, the connecting rod and the crank being in the same straight line, the effect of the piston pressure is to force the crank shaft against its bearings without rotating it. As the crank leaves its centers the arm of the piston pressure increases until the crank and connecting rod are at right angles (nearly), when, with a constant piston pressure, the turning moment would be a maximum. Equation 5 shows how the tangential pressure on the crank pin can be derived from the piston pressure. If, in Fig. 1, *OB* be drawn to represent to scale the pressure on the piston, *OH* will represent to same scale the tangential pressure on the crank pin when the connecting rod is in the position *AB*.

In Fig. 5 take the ordinates of the corrected diagrams and lay them off from *O* on the corresponding positions of the crank, and through the outer ends of these radial ordinates draw a line parallel to the corresponding position of the connecting rod, the intercepts on the vertical radius *OE* will give to scale the tangential pressure on the crank pin. In illustration of the foregoing, *Oa*, Fig. 5, is laid off equal to 1′1″ in the corrected diagram; through *a* draw *aH* parallel to the connecting rod 1 1″, then *OH* will represent to scale the tangential pressure when the crank pin is at 1, and so on for the other ordinates. The pressure which *OH* represents to scale is the total tangential pressure on the crank pin at the position 1, divided by the area of the piston. From these tangential pressures as ordinates may be constructed a tangential diagram, either on a straight line or a circular base. For an operation occurring in cycles the latter construction seems more appropriate. The crank pin circle may be taken for the base of

the ordinates, and from it lay off $1t = OH$, and so on for the other positions. Joining the outer ends of these radial ordinates gives the tangential diagram $DtGLD'ML'$. The tangential pressure for any point can be found by drawing a radial line through the point and measuring with the scale (1 in. = 40 lbs. in this case) the part of the line between the base circle and the curve of the tangential diagram.

The twisting moment is found by multiplying the tangential pressure at any point by the length of the crank; but as the length of the crank is a constant factor, the relation of the ordinates of the tangential diagram will be the same as that of the ordinates of the diagram of twisting moments. And for purposes of comparison the former diagram is all that is required.

The points where the corrected diagrams cut the zero line of pressures give the corresponding points where the curve of the tangential diagram passes inside the base circle, showing that the crank pin is dragging the reciprocating parts.

The curve of tangential pressure will always pass through the points $D$ and $D'$, for at the dead points the tangential pressure is always zero.

The distance $CD$ represents to scale the mean tangential pressure for one revolution, found by taking the mean unbalanced pressure from the diagrams in Fig. 4, and multiplying it by 2×stroke; this gives the work on the piston for one revolution, and must equal the work on the crank pin for one revolution. Equating these quantities of work, gives

$$P^m \times 2 \times 3 = P^{mt} \times 3 \times \pi.$$

$P^m = 33.65$ lbs. = mean unbalanced pressure of both diagrams of Fig. 4; therefore

$P^{mt} = 21.42$ lbs. = mean tangential pressure on the crank pin $= CD$ to scale. The circle $CBC'B'$ shows at a glance when the tangential pressure is equal to, greater or less than the mean tangential pressure for one revolution, and how much the former varies from the latter.

Were this engine used for driving a vessel, the propeller should be set on the shaft so that the blades would not pass the rudder post when the tangential pressure on the crank pin is at

its maximum, because the points of maximum pressure acting as impulses, and occurring at regular intervals, will produce vibrations whose amplitude will depend largely on the intensity of the impulse.

Of course this impulse is partially absorbed in accelerating the rotary motion, but the advantage still holds of having the blades in a position where the maximum reaction upon them will tend to force the vessel ahead, instead of being dissipated in producing wasteful impact on the stern and rudder posts.

With compound, triple or quadruple expansion engines the tangential pressure will be variable, even when the diagrams for all the pistons are combined, so that there are certain relative positions of cranks and blades that must be avoided, in order to reduce the vibrations of the vessel. For one of these points see Diagram of Crank Effort of Port Engine of U. S. S. *Yorktown*, by Messrs. McFarland and Norton, in Vol. I, No. 2, of the JOURNAL. This digression in regard to positions of cranks and propeller blades is indulged because of the practical tendency it gives to the foregoing graphical investigation.

The original indicator diagrams from an engine may be used directly by taking the half length for the length of the crank, which must be divided into as many equal parts as there are feet in the velocity of the crank pin per second, and then multiplying $P'$, equation 9, by the number of pounds found by the indicator scale in one of these divisions of the crank radius.

In practice, one figure would take the place of Figs. 2 and 5, which were used to avoid a confusion of lines in the explanation of the foregoing method. Many lines may be omitted, as two points of a line are enough for its direction, and this is what is required in many cases. There are some short cuts which need no explanation here, as they will suggest themselves as the different steps of the foregoing graphical method are taken.

[The dimensions in the plate have been reduced one-half.—ED.]

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ASSISTANT ENGINEER EMIL THEISS, U. S. NAVY.—As bearing on Mr. Canaga's paper, the following method of correcting indicator diagrams for the weight of reciprocating parts will, I think, prove of interest:

The inertia curves having been once constructed for different ratios of length of connecting rod to throw of crank, they may be made to answer for any engine, whatever its piston speed or weight of reciprocating parts, by simply reading off the ordinates on the proper scale. As a ready method of constructing these curves, the following table of ordinates for the more ordinary ratios of connecting rod to crank, adapted, together with the explanation, from an article on "The Successes with Fast-Running Engines," in the number of October 5, 1889, of *Die Zeitschrift des Vereines Deutscher Ingenieure*, will be found of service:

"The pressure due to the acceleration of reciprocating parts is given by the expression

$$\frac{Mc^2}{r}(\cos x \pm \lambda \cos 2x)$$

where $M$ = mass of reciprocating parts (best taken as so much per square inch of piston area).

$c$ = velocity of rotation in feet per second.

$r$ = length of crank in feet.

$x$ = the angle of rotation.

$\lambda$ is the ratio crank ÷ connecting rod.

The stroke is divided into ten equal parts, and the eleven ordinates calculated for the proper values of $x$, which are different for similar piston positions with different values of $\lambda$.

| Ordinates. | $\lambda = \frac{1}{4}$ | | $\lambda = \frac{1}{5}$ | | $\lambda = \frac{1}{6}$ | Ordinates. |
|---|---|---|---|---|---|---|
| 0 | [—]1.25 | | [—]1.2 | | [—]1.167 | 10 |
| 1 | [—]0.941 | Inward | [—]0.92 | Inward | [—]0.840 | 9 |
| 2 | [—]0.664 | | [—]0.639 | | [—]0.633 | 8 |
| 3 | [—]0.409 | | [—]0.379 | | [—]0.376 | 7 |
| 4 | [—]0.129 | | [—]0 126 | | [—]0.134 | 6 |
| 5 | 0.101[—] | | 0.091[—] | | 0.078[—] | 5 |
| 6 | 0.318[—] | | 0.300[—] | | 0.284[—] | 4 |
| 7 | 0.471[—] | Outward | 0.465[—] | Outward | 0.458[—] | 3 |
| 8 | 0.610[—] | | 0.613[—] | | 0.608[—] | 2 |
| 9 | 0.715[—] | | 0.717[—] | | 0.742[—] | 1 |
| 10 | 0.75 [—] | | 0.800[—] | | 0.833[—] | 0 |

These ordinates, being calculated for $\frac{Mc^2}{r} = 1$, must in each case be multiplied by the actual value of this expression.

For intermediate values of $\lambda$, the arithmetical means may be taken without serious error."

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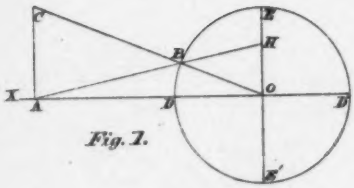
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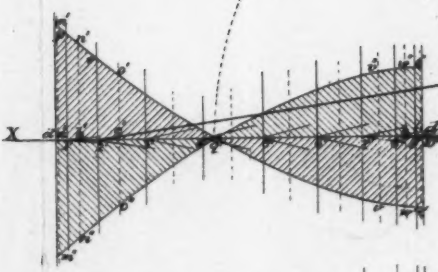
Fig. 1.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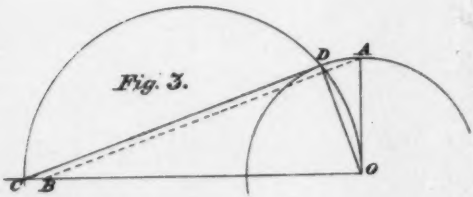


Fig. 3.
D
A
C
B
O

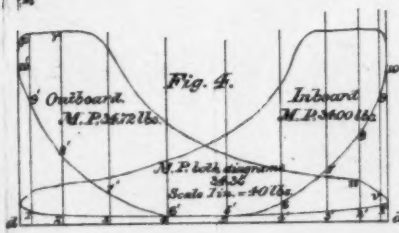
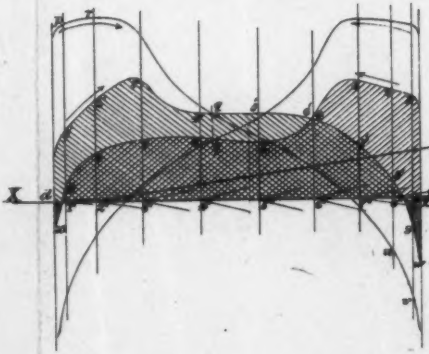


Fig. 4.
Outboard.
M.P.34172 lbs.
Inboard.
M.P.34001 lbs.
M.P. both diagrams
34.52
Scale 1 in. = 40 lbs.


X

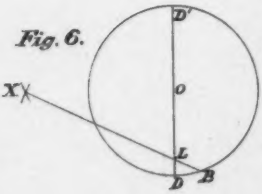


Fig. 6.
D'
X
O
L
D
B

*Fig. 2.*

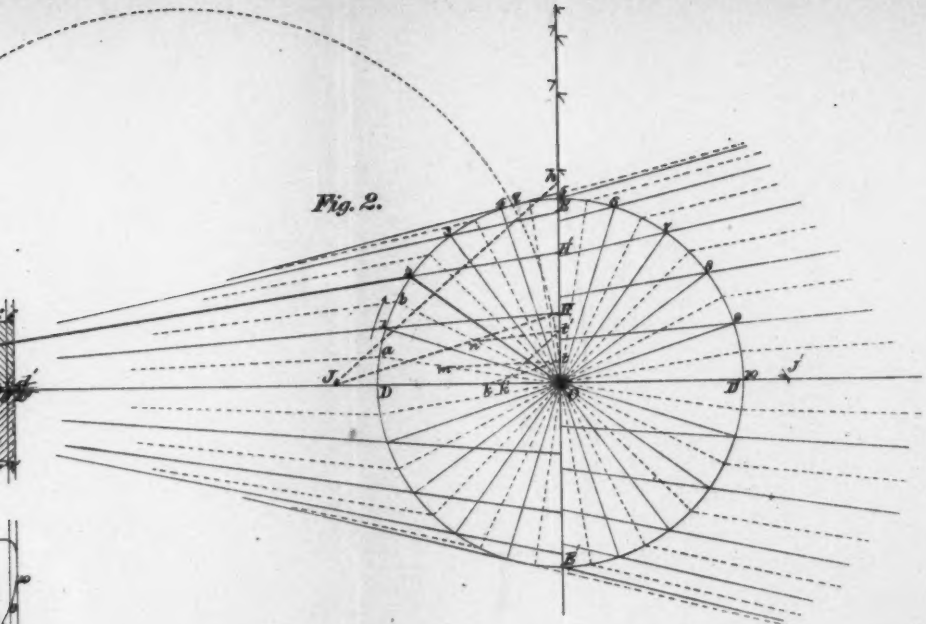
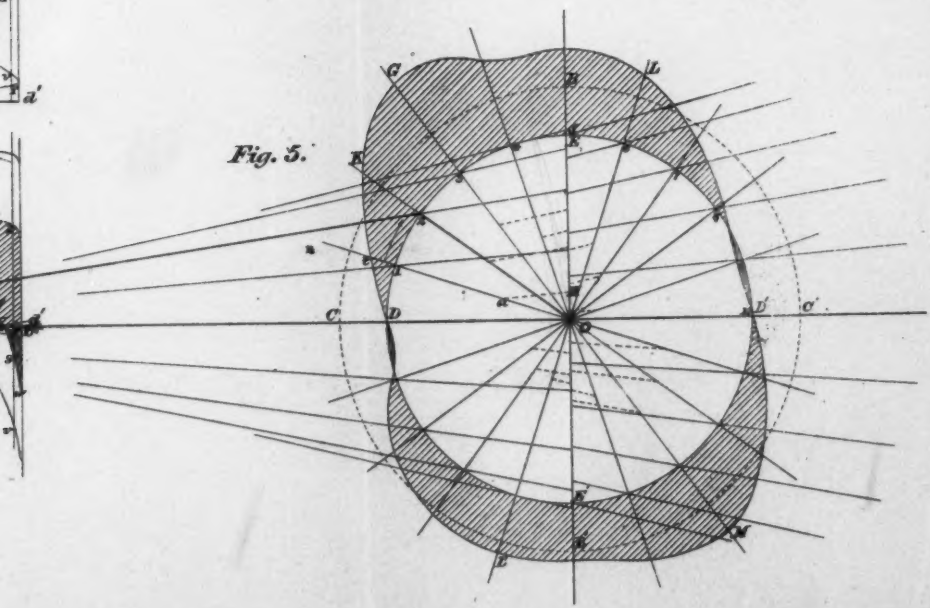


*Fig. 3.*



# XIX.

# EVAPORATORS.

By Passed Assistant Engineer G. W. Baird, U. S. Navy.

Not a great many years ago it was currently believed that the flat, insipid, oleaginous taste of freshly-distilled water was due alone to the absence of air; it was disputed that there was any oil in the water, and, indeed, it appeared difficult for an appreciable quantity of oil to reach the distiller on board ships when jet condensers were employed; it could not be disputed, however, that it was much worse on board vessels having surface condensers. It was well known that the water, after a number of days' exposure in the tanks, became sweet, potable, sharp and wholesome.

Proceeding on this belief, the writer devised* various means for accomplishing the aeration, believing that this was all that was needed to perfect the distiller. In this he was greatly mistaken, for after about five years of patient experimenting, and having accomplished the complete saturation of the water with air, he found himself but fairly entered on a field of research. Sound mechanical principles, proportions made with mathematical exactness and perfected by careful experiment, had done much, but it was found necessary, in order to effect a complete solution of the problem, to bring chemistry to the aid of mechanical science. A record of the experiments made by the writer at that time might amuse the well-informed engineer or chemist of to-day, but could serve no good purpose in these pages. It was not until a series of qualitative analyses were made by a good chemist that the writer was able to complete the little distiller.

Briefly, the oxygen of the air oxidized much of the solid and volatile matter which came over with the steam, but the oxidation was not thorough. A device for producing a current of oxy-

* Journal Franklin Institute, Vol. LXIII, p. 61.

gen gas, to be mixed with the air, was added, with better results. However, in one case this oxidation made the water taste much worse; the presence of the gas and of air had restored the "sharpness" to the water, but had introduced an empyreumatic odor and taste which were very disagreeable, due, the chemist said, to organic matter. To oxidize this more completely, the aerator and oxygenator were carried further from the condenser—*i. e.*, nearer to the boiler. This kept the steam and air longer in contact, and its good effect was at once apparent. The water thus made became potable in three or four days; *i. e.*, it precipitated during that time the solid matter it held in suspension. The writer, after some experimenting, found a filter which readily took up the oxides, and the result was a combination of the essential parts of a good distiller. The oxygenator, though beneficial, was abandoned, on account of the attention it required, the water being perfectly good in twelve or fifteen hours after distilling without its use.

At that time we were using mostly sperm and olive oils in our steam cylinders; we were employing low steam pressures, seldom more than 40 pounds above the atmosphere. If the boilers were clean and contained clean water the product of distillation was good, but if they did not the water was bad, but in time became good in the tanks. The tanks required frequent cleaning, and those on board vessels with surface condensers required much more cleaning than those on ships where jet condensers were employed. The reason is that the oil injected into the valve chests and cylinders in the former found its way to the boiler, and in some form or other to the distiller; its combination with other material in the water, its oxidation and precipitation appeared as dirt in the bottom of the tanks.

This grease, though the most abundant, is not by any means the only source of contamination.

When the double expansion engine, with its necessarily higher pressure was introduced, it was found that the animal and vegetable oils suffered a destructive distillation, and could, therefore, no longer be used to lubricate the valves or pistons. The so-called mineral oils—mixtures of some of the products of petro-

leum, among them paraffine with menhaden oil—made their appearance, and the effect of these upon the distilled water is what directed the writer's attention to an evaporator.

The essential qualities of a successful machine for this purpose are few: it must be light, of small volume; its heating surfaces must be easily removable for scaling; it must deliver its condensed water below the temperature of 212°, as it should drain into a feed tank to be pumped back into the boilers, and if it is above 212° there will be a revaporization and consequent loss; it should vaporize the sea water at as low a temperature as is consistent with economy and the purity of the output. There must be an excellent circulation to prevent priming.

The success of such a machine—indeed of most machines—depends almost entirely on its proportions. To proportion the different parts so as to yield the best results, and yet make the apparatus one easily managed by unskilled hands is not as easy as might be imagined. The proportions could neither be guessed at nor calculated. The writer, therefore, took up the subject by experimenting.

The first test was made with an improvised evaporator having a cast iron cylindrical shell and a single brass coil. A Chapman steam trap was used. Steam from the boiler was introduced into the bottom of the coil and discharged at the top. The upper head of the evaporator was left off. The feed was regulated by hand.

EXPERIMENT TO DETERMINE THE EFFICIENCY OF A COIL EVAPORATOR UNDER CONDITIONS OF ACTUAL PRACTICE.

| | |
|---|---|
| Duration of experiment | 2 hours. |
| Mean steam pressure in the coil, in pounds per square inch above the atmosphere | 20. |
| Mean temperature of the steam in the coils in degrees Fahrenheit | 259.3 |
| Mean temperature of the feed water in degrees | 131.66 |
| Total number of pounds of water vaporized | 206.66 |
| Total number of pounds of water condensed in the coils, and delivered by the steam trap | 224.25 |
| Mean temperature of the water vaporized | 212. |
| Total heat in the steam in the coils | 1,193.7 |
| Total heat in the steam in the evaporator | 1,178.6 |

Quantity of heat destroyed in the steam condensed in the coil [1193.7 — 259.3] × 224.25 = ..... 209,539.2

Quantity of heat accounted for in the vaporized water [1178.6 — 131.6] × 206½ = ..... 216,373.

Pounds of water vaporized per hour $\left[\frac{206.66}{2}\right]$ = ..... 103.33

Pounds of water vaporized per hour per square foot of heating surface $\frac{103.33}{4.876}$ ..... 21.180

Thermal units transmitted per hour per square foot of heating surface.... 22,178.

Thermal units transmitted per minute per square foot, per degree difference of temperature $\frac{22178}{60[259.3 - 212]}$ = ..... 7.81

I repeated this experiment, and obtained sensibly the same quantities. I submitted my figures to Mr. Isherwood, and begged that he advise me why the thermal units destroyed inside the coil did not balance those produced outside that surface, and his answer was that part of the water condensed inside the coil probably drained back into the boiler, whence the steam was supplied.

As the water foamed violently in this little evaporator I built up on the height of the shell about 30 inches, to prevent the water foaming over. I repeated the experiment with low water in the evaporator, and continued to lower that level until the foam barely covered the top of the coil, and found I got practically the same vaporization as when solid water covered the coil. The height to which the water would foam was found to be about 15 inches.

This is a verification of an experiment made many years ago by Mr. Isherwood on a vertical tubular boiler. A progressive series of tests was made to determine the economic efficiency of water heating and steam heating surfaces. After a test had been made with the water at its normal level subsequent tests were made with the water one inch below the top tube sheet; the next with the water two inches below, and so on until the water was fifteen inches below the top tube sheet, or with half of the length of the tube above the water level. There was no difference in either the potential or the economic vaporization of the boiler, nor was there any indication of over-heating of the metal nor superheating of the steam until fifteen inches had been reached. Mr. Isherwood accounted for this then curious fact, by reasoning that the foaming of the water reached the top of the tube and

kept it wet, and also that the water in the lower half of the brass tube, being a more powerful absorbent of the heat than the steam, was sufficient to receive the heat transmitted.

The test on the vertical tubular boiler, however, was with a low rate of combustion (about ten pounds of coal per square foot of grate per hour), in which case the rate of transmission of heat per square foot of heating surface was very much less than in the evaporator test; indeed, the transmission of heat in this test, with but twenty pounds of steam in the evaporator coil, was more than three times as great as that in the boilers of the *Yorktown* during her test under a forced blast.

By extending experiments the writer found that, by using higher pressures, higher potential results were obtained, but the water was delivered to the trap at a higher temperature. This could be reduced by lengthening the coil and diminishing its pitch, which was done in subsequent machines.

The results obtained in these experiments are not so great as those reported from abroad, of which, however, the writer had no knowledge at that time. The close agreement of my own successive tests is a guarantee of their accuracy.

### Experiment to Determine the Efficiency of a Number OO Baird Evaporator having an Iron Coil.

The writer then directed his attention to the efficiency of an evaporator with a single iron coil, of commercial ¼-inch pipe, as shown in Fig. 2, page 295. It contained 1.34 square feet of heating surface, measured on the outside of the coil:

| | |
|---|---|
| Duration of test, in hours and minutes | 10–38 |
| Mean steam pressure in the coil, in lbs. above the atmosphere | 35. |
| Mean pressure in the evaporator above atmosphere | 0. |
| Mean temperature of the feed water | 76. |
| Mean temperature of steam in evaporator | 212. |
| Mean temperature of steam in the coil | 281. |
| Total heat in the steam in the coil | 1,199.1 |
| Total number of pounds of water vaporized | 250. |
| Mean number of pounds of water vaporized per hour | 23.5 |
| Total number of gallons of water vaporized per hour | 30.12 |
| Mean number of gallons of water vaporized per hour | 2.833 |

| | |
|---|---|
| Mean number of gallons of water vaporized per hour per square foot of heating surface | 2,114. |
| Quantity of heat accounted for in the water vaporized per hour, (1178.6 — 76) × 23.5 = | 25,910. |
| Mean number of pounds of water vaporized per hour per square foot of surface | 18.3 |
| Thermal units transmitted per hour per square foot of heating surface | 19,336 |
| Thermal units transmitted per minute per square foot of heating surface per degree difference of temperature, $\frac{19336}{60[281-212]}$ = | 4.67 |

This result is less than that obtained from the brass coil, though the former was No. 18 B. W. G. in thickness, while the latter coil was about No. 14. They were both single coils, and probably drained equally well. These tests were with clean coils. Of course their efficiencies would be affected very sensibly by a coating of boiler scale, which they are certain to get, and for this reason the writer allows additional surface in practice.

In this last experiment the glass water gauge fittings were tapped into the side of the evaporator. The shell was filled until the water rose to about the middle of the glass. As soon as steam was admitted to the coil, there appeared alternately a globule of water and a bubble of steam which flowed, in rapid succession, through the glass from bottom to top so long as there was sufficient water in the shell to reach the lower fitting of the gauge glass. This apparent indication of foaming, though pretty to look at, was most provoking, as it proved the unreliability of my water gauge. I then connected the gauge glass from the top and bottom of the shell, as shown in the engraving, when the water became perfectly steady in the glass. Permitting the pressure to accumulate in the shell and suddenly opening the safety valve and allowing the pressure to fall made no sign of motion in the glass. The writer was, therefore, satisfied that no serious foaming existed.

The No. 2 plant for the *Petrel*, the No. 3 for the *Yorktown*, the *Concord* and the *Bennington*, and the pair of No. 4 for the *Charleston*, were tested by Chief Engineer McKean. These vessels have "complete plants," *i. e.*, evaporators, aerators, condensers and filters, and steam traps. These, with a No. 6 Davidson L. S. circulating pump, were erected in the machine shop of the

Davidson Steam Pump Company, at Brooklyn. The pump was larger than the one belonging to the plant, but was run at a lower speed. After ascertaining that all the parts called for had been made, were of good quality, and were ready for delivery, Mr. McKean proceeded to make three kinds of tests. He required a pressure to be exerted in the shells, to see that joints, rivets, seams, etc., were tight. He made a test for capacity, using the highest pressure the boiler in the shop afforded, and a test for quality, running the evaporators at about their rated capacity. The feed water used was the Ridgewood (Brooklyn City) hydrant water with a quantity of Turk's Island salt dissolved in it, the purpose being to detect foaming, for if any water primed over it would be detected in the distilled water by the solution of nitrate of silver which was provided. The solution was much stronger than sea water.

The following is a copy of the tabulated results:

| Size of machine. | No. 3 Plant. | | No. 2 Plant. | |
|---|---|---|---|---|
| Evaporating surface | 96 sq. ft. | | 46.3 sq. ft. | |
| Condensing surface | 34 sq. ft. | | 23 sq. ft. | |
| Duration of test while taking data for averages. | Test for quantity—two hours. | Test for quality. | Test for quantity—one hour and ten minutes. | Test for quality. |
| Steam pressure in the coils | 49 | 55 | 41.8 | 47 |
| Steam pressure in the evaporators | 23.5 | 41 | 30.5 | 46. |
| Temperatures: Injection water | 76. | 76 | 73 | 74 |
| Temperatures: Discharge water | 87.3 | 86 | 84 | 83.5 |
| Temperatures: Distilled water | 95.5 | 84 | 81.25 | 78. |
| Strokes of circulating pump per minute | 112 to 120 | 120 | 93. | 108. |
| Size of pump cylinder | $6\frac{1}{2}'' \times 12''$ | $6\frac{1}{2} \times 12$ | $6\frac{1}{2} \times 12$ | $6\frac{1}{2} \times 12$ |
| Gallons of water per day (rate) | 2845. | 2006. | 1731. | |
| Rated capacity | 2000 | ..... | 1000 | |
| Salt | Trace. | None. | None. | None. |

"In a portion of the trial," says the report, "carrying the

water at the height of three-fourths of the coils, there were slight traces of salt in the water produced, evidently carried over by foaming. With a less height nitrate of silver showed no trace of salt. As the inflow of steam has since been divided so as to make it pass through four branch pipes (dry pipes), and as it is intended to carry the water low in practice, according to 'directions,' no trouble is anticipated on that score."

Though a greatly less pressure was used in the coils than the running pressure which will be allowed on board the ships where these machines will be used, it will be observed that both sizes overran very much. It must also be remembered that these tests were with clean coils. After scale forms on the coils their power of transmission will be diminished. The temperature of the distilled water in these tests was very satisfactory; in three out of the four tests it was lower than the discharge water. This, at first, seems paradoxical, but when it is remembered that the distiller (condenser) coils have but little pitch, being single-threaded helices, and that their condensing surface is so large that the steam is condensed before it reaches half the length of the coil—the lower part (more than half) being utilized in cooling the water—also that the coldest circulating water is put in at the bottom of the condenser, one will readily see how and where the cooling is done.

While this is eminently proper for a distiller, it would be wrong for an engine condenser; and if, in such a condenser, it should be determined to use inside condensation, the coils should be in double or triple helices—*i. e.*, like a double or a triple threaded screw; this would make a larger number of shorter coils, having about the same aggregate surface.

The coils in the distillers tested by Mr. McKean had been previously tested (just after being bent) at 300 pounds per square inch, internal pressure. After reaching the Davidson factory they were tested at 160. Mr. McKean's test, which was external to the coils, but internal to the condenser, was 57 pounds. The only leak discovered was "a slight weeping about a few of the flange bolts" of one of the shells, "which were marked and made good."

Following are the net weights of the three classes:

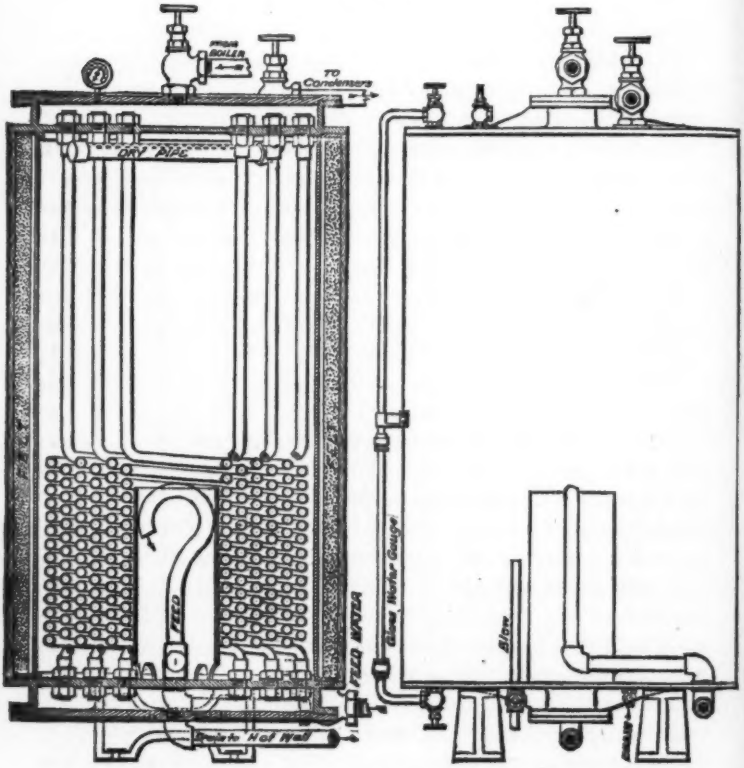
| | *Yorktown* and class. | *Baltimore.* | *Petrel.* |
|---|---|---|---|
| Condensers and fittings, . . | 585.5 | 1171. | 510.5 |
| Evaporators and fittings, . . . | 1493. | . . . | 1185. |
| Filters and charcoal, . . . . | 161. | 235. | 140. |
| Total, . . . . . | 2239.5 | 1406. | 1835.5 |

The evaporator of the *Yorktown*, it appears, made ($\frac{2845}{96\times24}$=) 1.24 gallons of water per hour per square foot of heating surface under the trial conditions. It will do in actual service, no doubt, quite as well, for the engineer will have, at his command, a greater range of temperature. On board the *City of Birmingham* there is a No. 5 Baird evaporator which has 174 square feet of surface in *iron* coils. It vaporizes 250 gallons an hour, or 1.44 gallons per hour per square foot of surface, with 160 pounds in the coils. The evaporators supplied by the writer to mercantile vessels have iron coils. As the coils soon become coated with scale, and as iron will stand the blows of a scaling hammer better than copper or brass the writer recommends them.

Fig. 1 shows the No. 4 evaporator, the first pair of which was purchased by the Union Iron Works for the *Charleston*. They each contain six copper coils, tinned. Each coil has 14 convolutions, with no joint except at the ends. The total heating surface in each one is 108 square feet. The shell, which is 28″ in diameter and 54″ in height, is made of a single plate of ingot iron ("low steel") with the longitudinal seam welded and the ends flanged. The heads are of cast iron and are from the same pattern. Each head contains a steam chest into which all the coils open. The feed water is delivered near the top of the central chamber, descends and reaches the coils from below. The top of the water should be carried about 18″ below the top coil (about the middle of the glass). The water condensed inside the coils goes to the steam trap, thence to the feed tank. Steam from main boiler is supplied to the top steam chest. Steam gauges show the pressure in the coils and in the evaporator. In running

the machine care should be taken to avoid getting the drain water above the boiling point, and to keep the water low in the glass, *i. e.*, not above the middle of the glass ordinarily. The density of the water will change rapidly, as the vaporization is rapid and quantity of water small; but a little experience will enable

Fig. 1.



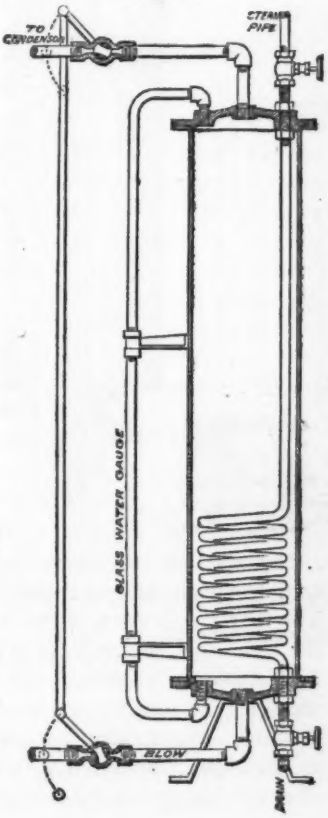

the operator to set a steady blow and feed. It will be found that, with an independent steam feed pump, the supply of feed will not be difficult to regulate.

The quantity of water condensed in the coils is large, and the

ordinary steam traps, which have small outlets, will not answer for this purpose. The trap must be a fairly close regulator, and must have a large outlet, as the drain water, under the high pressures now used, comes to the trap sometimes intermittently and with violence.

There is a salinometer pot with each of the larger evaporators, and as the amount of water vaporized is very large in proportion to the amount contained in the shell, the density of the water will require to be watched unremittingly. It will be about one man's work to attend a pair of evaporators, condenser and circulating pump when they are running at their normal capacity.

Fig. 2.




For small vessels, and especially for launches, the style shown in Fig. 2 is used. To avoid the necessity of a salinometer, the apparatus has been so proportioned that if vaporization commence when the gauge-glass is just filled with normal sea water the saturation will be about $\frac{6}{32}$ when vaporized down to the bottom of the glass.

The top cock is connected to the condenser and the bottom one to the sea. Both cocks are operated by the one rod, and are so placed that one position of the handle will open only the top one, another position will open only the bottom one, and a third position will open both. Hence it follows that if in the third position the sea water will be drawn in (fed), in the first posi-

tion vaporization will take place and the vapor pass into the condenser, and in the second position (where the top one is closed and the bottom one open,) steam pressure will form and blow the water out of the shell. The handle has stops, and no time need be consumed in adjusting its position.

Fig. 3.

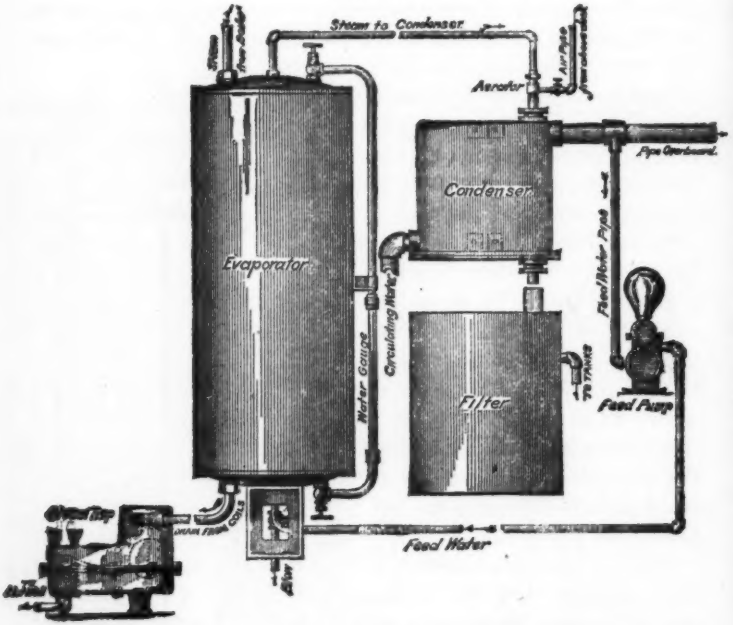



Fig. 3 shows the assembling of the several inherent parts of a complete plant, excepting that the circulating pump is not shown. The feed pump takes its water from the outboard discharge of the condenser, which is usually ten or twelve degrees warmer than the injection water. The temperature may be varied by changing the speed of the circulating pump.

It might appear that the evaporator would be less economical to distil from than a donkey boiler, but such is not the fact; for after the donkey boiler has become scaled, the heat of the furnace,

instead of being conducted through the metal into the water, escapes, in larger and larger quantities, up the chimney. With the evaporator the case is very different, for no heat disappears inside the coils, but is absorbed by water on the outside of the coils. The effect of scale on the coils of an evaporator retards the action of the machine, but does not diminish the economic performance except in the slightly greater ratio of external radiation, compared with the internal transmission; but this ratio is much less than in a boiler, as its radiating surface is less.

Continuing the comparison, the evaporator is much lighter than a boiler of equal power; it occupies less space; it is more cleanly, and it is safer.

In erecting the evaporators care should be taken to select a place where the height is sufficient for the removal of the coils. The holding-down bolts in the feet, and what bracing the pipes afford, will be sufficient to hold them. An automatic pressure regulator fitted in the live steam pipe from the boiler will hold the pressure within the limit determined upon, and will regulate the supply of heat to the evaporator, and consequently its output. It will also act as a safety valve, as the evaporator pressure can never exceed the coil pressure. Care should be taken that flange unions be placed in all the pipes on the upper head and in the drain on the lower head, to provide for easily breaking and making joints. A spare set of coils should be provided for each vessel. In a merchant steamship these may be left on the company's wharf, ready to replace the set in use when the latter requires scaling, but as the war vessel makes her home on the seas, she should carry the spare coils. By this means a nest of clean coils may be kept in readiness.

To *remove* the coils for scaling, it is necessary to remove both steam chest bonnets and take off the outer coil nuts. Nothing will be saved by keeping the top nuts fast and hoisting all the coils out together, for they must be separated to be scaled. It is, therefore, better to remove the nuts and take the coils out separately, commencing with the inner coil. It will be found that the springing of a coil in handling it will crack much of the scale off, and that light scaling hammers will readily complete the process. In replacing the coils, commence with the outer one.

The steam supply to the pumps should come from the live steam pipe beyond the pressure regulator, and their exhaust should go to the auxiliary condenser; the valve (if there is one) in the steam pipe from the evaporator to the distiller should be run wide open, and the pressure in the evaporator governed by the supply of live steam. The circulating pump should have no bilge connection whatever.

To *test* for tightness, empty the evaporator, open the drain cock under the shell and admit live steam to the coils; if there is a leak steam will issue from the drain cock.

To *test* the condenser, run the circulating pump without letting any steam into the coils; if there is a leak in either coil or manifold it will show in salt water dripping from the fresh water discharge. By closing the coil valves, alternately, the leaky coil can be determined. If the condenser has no coil valves, and if it is imperative to continue distilling, the leak may be reversed by partially throttling the fresh water outlet under the condenser; this, it will be seen, causes an accumulation of pressure on the fresh-water side and makes a leak the other way, but enables fresh water to be made for the time.

Where sufficient height will not be given for placing the evaporator for removing the coils, the shell is made in halves. By removing the upper half the coils may be taken out, scaled and replaced without disturbing the lower half of the shell, but it is much better to obtain sufficient room in the first place for so important a machine. When it is considered what an immense saving of weight a distilling plant effects on board a ship there should be no question about a few cubic feet for its accommodation, nor a few feet for the removal of its coils. It is quite within the recollection of chief engineers now on the active list for ships of the Navy to carry 120 tons of water in their tanks, whereas vessels of the same size now carry about one-fifth of that amount. The present distilling apparatus weighs less than one-fifth of the iron tanks formerly carried for the storage of the water.

## XX.

# AN ACCOUNT OF AN EXPERIMENT MADE BY MESSIEURS WM. ROTCH, WM. R. BILLINGS AND R. C. P. COGGESHALL, ON THE TWO VERTICAL CORLISS BOILERS FURNISHING STEAM TO A WORTHINGTON DIRECT ACTING, HORIZONTAL, COMPOUND, CONDENSING PUMPING ENGINE SUPPLYING THE CITY OF NEW BEDFORD, MASSACHUSETTS, WITH POTABLE WATER, TO DETERMINE THEIR ECONOMIC VAPORIZATION OF WATER BY LEHIGH ANTHRACITE.

By Chief Engineer B. F. Isherwood, U. S. Navy.

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The experiment described in the following pages was made by three experts acting under the order of the New Bedford Water Board, on the two new vertical Corliss boilers furnishing steam to the new Worthington pumping engine supplying potable water to that city.

The three experts were Messieurs Wm. Rotch, Wm. R. Billings and R. C. P. Coggeshall. Their report to the Water Board will be found in the eighteenth annual report (1888) of that board to the City Council of New Bedford, page 24. The object of the experiment was to determine the economic vaporization of water by the boilers with Lehigh anthracite when the engine was pumping at its normal duty. From the information in that report, kindly supplemented by Mr. Coggeshall, this paper has been prepared.

### BOILERS.

The boilers were two in number, and of the type frequently employed by Corliss to furnish steam for his pumping engines.

Each boiler consisted of an iron cylindrical shell placed vertically, the lower end of the shell being occupied by a furnace con-

centric with it and separated from it by a water-leg. The top of the furnace was flat, and was also the lower tube plate of the tubes, which latter occupied the shell from the furnace top to the flat top of the shell, that top being also the upper tube plate.

Each shell was surrounded concentrically by a vertical wall of brick masonry, which extended sufficiently below the bottom of the shell to form the ashpit of the furnace. A narrow air space separated the outside of the boiler shell from the inside of its encircling wall of masonry. The portion of the wall surrounding the ashpit is twice the thickness of the remaining portion, a square offset being made on the inside, beneath the bottom of the boiler shell to support the latter. There is riveted to the boiler shell near its top, a little above the top of the brick wall, an annular ring of iron, which extends horizontally over the air space and the wall. To the outer circumference of this ring a vertical piece of sheet iron is secured, which, projecting downwards as a lip, shuts over the top of the wall like the lid of a cannister, the object being by closing over the air space to prevent the circulation of air within it, and to allow the shell to move vertically with changes of temperature without danger of dislocating any part of the structure.

The purposes of the wall of brick masonry surrounding the boiler shell were to prevent the loss of heat by conduction and radiation, to form the ashpit beneath the furnace, and to support the boiler shell. The opening for the furnace door was through both the masonry and the water leg surrounding the furnace. The pipes leading from the shell to the feed valve, blow-off valve, water gauge, etc., of course penetrated the brick wall.

The grate was circular, and of the kind known as Bannister's rocking grate. The bottom of the grate was a little above the bottom of the shell.

On the top of the horizontal iron ring covering the brick wall and air space was a sheet iron uptake, cylindrical, with a flat top, and from the side of this uptake a rectangular cross-sectioned sheet iron flue carried the gases of combustion to the chimney. The diameter of the uptake was the same as the inner diameter of the wall, and all the boiler tubes discharged directly into it.

Upon the upper tube plate of the boiler and wholly within the uptake was a small steam drum concentric with the boiler shell. Of course, there were no tubes immediately beneath this drum, which covered about 17⅓ per centum of the area of the tube plate; consequently, there was an unbroken water space of this area surrounding the axis of the shell concentrically, which formed, in connection with the water space between the outer row of tubes and the shell, ample room for the return or descending currents of water, the ascending ones being caused by the heat absorbed from the tubes.

From one side of the steam drum a steam pipe was led horizontally through the uptake; and on it, outside of the latter, was placed the safety valve. The stop valve was placed in a horizontal chamber bolted to the exterior end of this pipe, and from the side of the chamber the steam pipe was continued to the engine.

The boiler tubes were of iron, and extended vertically between the top of the furnace and the upper end of the shell which formed the tube plates. The gases of combustion were within the tubes. Of the total length of the tubes five-eighths were surrounded by water and the remaining three-eighths by the steam, so that of the total tube surface five-eighths were employed in generating steam and three-eighths in superheating it. The surfaces of the steam drum, and of the steam pipe within the uptake, and of the upper tube plate, were also steam superheating surfaces. On the top of the uptake, and covering the area occupied by the tubes, is a removable cast iron plate annular in surface. This plate has to be removed whenever the tubes are to be swept.

The following are the principal dimensions and proportions of ONE boiler.

| | |
|---|---|
| Exterior diameter of the brick wall | 6 feet 4 inches. |
| Height of the brick wall from bottom of ashpit | 14 feet 6 inches. |
| Thickness of the brick wall at the ashpit | 9 inches. |
| Thickness of the brick wall above the ashpit | 4½ inches. |
| Height of boiler from bottom of ashpit to top of uptake | 16 feet 9 inches. |
| Diameter of the boiler shell | 5 feet 4 inches. |
| Height of the boiler shell | 13 feet 7 inches. |

| | |
|---|---|
| Width of air space surrounding boiler shell | 1¼ inches. |
| Height of steam drum on boiler shell | 1 foot 2 inches. |
| Diameter of steam drum on boiler shell | 2 feet 6 inches. |
| Diameter of the uptake | 5 feet 7 inches. |
| Height of the uptake | 1 foot 2 inches. |
| Diameter of the ashpit | 4 feet 10¾ inches. |
| Height of the ashpit to bottom of shell | 1 foot 6¾ inches. |
| Diameter of the furnace | 4 feet 10¾ inches. |
| Height of the furnace above bottom of shell | 2 feet 7 inches. |
| Width of water leg surrounding furnace, including thicknesses of metal | 2.8125 inches. |
| Height of opening of furnace door | 1 foot 2 inches. |
| Cross area of flue (3½ by 1 foot =) connecting with chimney | 3.5 square feet. |
| Horizontal length of flue connecting with chimney | 86 feet. |
| Height of chimney above grate surface | 70 feet. |
| Number of tubes in boiler | 128. |
| Outside diameter of tubes | 2 inches. |
| Inside diameter of tubes | 1¾ inches. |
| Length of tubes to outside of tube plates | 10 feet. |
| Length of tubes in water, including lower tube plate | 6 feet 3 inches. |
| Length of tubes in steam, including upper tube plate | 3 feet 9 inches. |
| Inside diameter of steam pipe | 5 inches. |
| Length of steam pipe in uptake | 1 foot 6 inches. |
| Area of the grate surface | 18.5858 square feet. |
| Aggregate cross area of the tubes for draught | 2.1380 square feet. |
| Aggregate area of water heating surface in the tubes, calculated for their inside circumference | 364.0765 square feet. |
| Area of water heating surface in the top of the furnace | 16.4478 square feet. |
| Area of water heating surface in the sides of the furnace | 49 4419 square feet. |
| Total area of water heating surface in the boiler | 429.9662 square feet. |
| Aggregate area of steam superheating surface in the tubes, calculated for their inside circumference | 217.4685 square feet. |
| Area of steam superheating surface in top of shell | 14.6008 square feet. |
| Area of steam superheating surface in steam drum | 9.1630 square feet. |
| Area of steam superheating surface in steam pipe | 1.9635 square feet. |
| Total steam superheating surface in the boiler | 243.1958 square feet. |
| Square feet of water heating surface per square foot of grate | 23.134124 |
| Square feet of steam superheating surface per square foot of grate | 13.085033 |
| Fraction of a square foot of steam superheating surface per square foot of water heating surface | 0.565616 |
| Square feet of grate surface per square foot of cross area of the tubes for draught | 8.693078 |
| Water space in boiler | 126.5 cubic feet. |
| Steam space in boiler shell | 69.6 cubic feet. |
| Steam space in steam drum | 4.2 cubic feet. |

Steam space in steam pipe in uptake ..... 0.2 cubic feet.
Total steam space in boiler ..... 74.0 cubic feet.

## HEATER.

The heater in the flue for heating the feed water previous to its entrance into the boilers, and with the heat remaining in the gases of combustion after they have left the boilers, was in common for both boilers. It was composed of 700 feet length of *brass* pipe 1⅜ inches in outside diameter and 1 inch in inside diameter. The external surface of this pipe was 251.9825 square feet, and its internal surface was 183.26 square feet.

The fraction which the external surface of the heater was of the water heating surface in the *two* boilers is 0.2932. The fraction which the internal surface of the heater was of the water heating surface in the two boilers is 0.2131. The latter is the proper figure for comparison.

Considered in proportion to its surface, to the temperatures of the feed water and gases of combustion entering it, and to the quantity of water passing through it per unit of time, the economic efficiency of the heater was very great, as appears from the results of the experiment hereinafter given.

## MANNER OF MAKING THE EXPERIMENT.

On the morning of February 16th, 1887, the two boilers were supplying steam to the Worthington engine, which was then pumping at its normal speed. At 8.30 A. M. the engine was stopped, the fires drawn from the furnaces and the ashpits cleaned. At 9.12 A. M. new fires were started in each boiler.

At 9.30 A. M., when the experiment was commenced, the steam pressure in the boiler was 98 pounds per square inch above the atmosphere. At 9.54 A. M. the engine was started and ran continuously until 10.15 A. M., February 17th, 1887, the boiler pressure at that time being 111.5 pounds.

The height of the water in the boilers was carefully noted at the beginning and end of the experiment, and the proper correction was made for the very slight difference.

The fuel used was Lehigh anthracite.

The gauges and thermometers employed had all been previously tested. The barometer was an aneroid which had also been carefully tested.

The quantity of water fed to the boilers was measured in the following manner: a tank was placed upon a tested platform scale and supplied by the feed pump with water from the hot well of the engine, the temperature of the water being noted at the time of weighing. The water, after being weighed, was allowed to flow into a lower tank, from which it was pumped by a second feed pump into the boilers.

Two boilers were simultaneously in use.

Observations were taken every hour of the atmospheric pressure near the boilers, and of the following temperatures, namely: of the superheated steam at the boilers, of the feed water before and after its passage through the heater, of the gases of combustion in the flue before and after their passage through the heater, of the external air and of the air in the boiler room.

Observations were taken every half-hour of the steam pressure in the boilers, of the height of the water in the boilers, and of the quantity of water fed to the boilers.

The heater consisted of brass tubes of 183.26 square feet of internal surface placed in the flue common to both boilers.

The weight of anthracite given as consumed, is its gross weight as thrown into the furnaces, including its moisture. The weight of refuse from this anthracite includes not only the ash and clinker, but the unconsumed small pieces of coal which fell between the grate bars into the ashpit. It does not include the hygrometric moisture. The weight of combustible given as consumed is the gross weight of the anthracite less the gross weight of the refuse, the latter being weighed in the perfectly dry state.

Owing to radiation and the considerable inleakage of the external air to the uptakes and flues of the boilers, the temperatures of the gases of combustion before and after passing the heater are very much too small, but the difference may be taken as correct and as showing the amount of heat absorbed by the water passing through the heater.

The manner of ascertaining the temperature of the steam in the steam pipe at the boilers and at the engines was by thermometers, whose bulbs and part of whose stems were immersed in mercury contained in iron cups projecting downwards into the steam pipe nearly across its diameter.

The correct economic vaporization given by the boilers is, of course, the vaporization from the temperature of the feed water entering them; that is, of the feed water after leaving the heater.

When the engines are in operation, the water of condensation from the steam jackets of their cylinders drains into a closed tank beneath them, from which it is pumped by a small special steam pump directly into the boilers. As the jackets are supplied with steam through a special pipe, the pressure in the jackets may be assumed to be the boiler pressure, and, consequently, the water of condensation in the jackets will have the same temperature as the water in the boiler, the heat taken out of the steam in the jackets being only its latent heat. This water of condensation from the jacket was not measured during the experiment, but was measured on another occasion, with all the conditions the same, during eight consecutive hours, the discharge from the pump being received into a closed tank and weighed. The average for the eight hours was 286 pounds of water per hour, which have been taken as the quantity drawn from the jackets during the experiment and delivered directly into the boilers. This quantity is in addition to the number of pounds of feed water drawn from the hot well and separately weighed during the experiment, as above described.

The air pump is independent, and the exhaust steam from the small cylinder driving it is delivered into the hot well. The steam cylinders of the two independent feed pumps also exhaust into the hot well. The steam cylinder of the pump, drawing the water of condensation from the steam jackets, likewise discharges into the hot well. The hot well thus receives, in addition to the discharge of the air pump, the exhaust steam from four small cylinders driving pumps independently of the main engines.

The temperature of the water discharged from the condenser

into the hot well was, during the exhaust, 83 degrees Fahrenheit. The effect of the exhaust steam discharges from the independent or auxiliary engines was to raise this temperature to 123.27 degrees Fahrenheit, which was the temperature of the feed water when weighed as drawn from the hot well. With this temperature the feed water entered the heater, receiving therein sufficient heat to increase its temperature to 165.48 degrees Fahrenheit, with which it entered the boilers.

In the following table will be found the data and results of the experiment.

Under the head of "vaporization" there is given the total number of pounds of feed water that would have been vaporized in the boilers had it been supplied at the temperatures of 100 degrees and of 212 degrees, instead of at the experimental temperature of 165.48 degrees, and had it been vaporized under the standard atmospheric pressure instead of the experimental pressure of 129.53 pounds per square inch above zero. These quantities are obtained from the experimental vaporization by multiplying the latter by the number of Fahrenheit units of heat imparted to the water and dividing the product by 1078.5331 in one case and by 965.7 in the other case. The quotients are correct, on the supposition that the economic vaporization by the boiler is independent of the steam pressure, or rather of the water temperature, in it, which is not the case, as that temperature materially affects the result, as will be appreciated from the following considerations:

Evidently the economic vaporization depends, other things equal, on the temperature in the chimney to which the gases of combustion are reduced. The lower that temperature the higher the economy. Now, the lowest limit of temperature in the chimney of these gases is the temperature of the water in the boiler, and as the temperature of steam follows its pressure, the lower the pressure the lower will be that temperature. All boilers, under any conditions, can be made, by giving sufficient heating surface, to reduce the temperature of the gases of combustion to that of the water within, in which extreme case the fraction of the total heat of combustion that can be utilized is what remains of the temper-

ature of combustion after deducting the temperature of the water and steam in the boiler.

In the experimental case, the difference between the temperature of the water in the boiler (346.81 degrees Fahrenheit) and the temperature of 212 degrees is 134.81 degrees, or ($\frac{134.81 \times 100}{1800}$ =) 7.49 per centum, consequently the vaporizations given in the table should be increased in the ratio of (1.0000—0.0749=) 0.9251 to 1.0000 for strict accuracy under the assumed conditions.

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENT MADE ON THE TWO CORLISS BOILERS OF THE NEW BEDFORD WATER WORKS.

TOTAL QUANTITIES.

| | |
|---|---|
| Date of the experiment | February 16 and 17, 1887. |
| Duration of the experiment, in consecutive hours and minutes | 24.45 |
| Total number of pounds of anthracite consumed | 7780. |
| Total number of pounds of refuse from the anthracite | 1066. |
| Total number of pounds of combustible consumed | 6714. |
| Per centum of the anthracite in refuse | 13.7018 |
| Total number of pounds of water drawn from the hot well | 61,720.4 |
| Total number of pounds of water of condensation from steam jackets | 7,078.5 |
| Total number of pounds of feed water vaporized in the boilers | 68,798.9 |
| Per centum which the water of condensation from the steam jackets was of the water vaporized in the boilers, by weight | 10.289 |

PRESSURES.

| | |
|---|---|
| Atmospheric pressure in inches of mercury | 29.64 |
| Atmospheric pressure in pounds per square inch | 14.55 |
| Steam pressure in boilers in pounds per square inch above the atmosphere | 114.98 |
| Steam pressure in boilers in pounds per square inch above zero | 129.53 |

TEMPERATURES IN DEGREES FAHRENHEIT.

| | |
|---|---|
| External air | 36.96 |
| Air in the boiler room | 81.12 |
| Water in the boiler | 346.81 |
| Steam in the boiler | 369.04 |
| Number of degrees the steam was superheated | 22.23 |
| Gases of combustion before passing the heater | 366.04 |
| Gases of combustion after passing the heater | 262.64 |

| | |
|---|---|
| Feed water, exclusive of the feed water of condensation from the steam jackets, before entering the heater | 123.27 |
| Feed water, exclusive of the feed water of condensation from the steam jackets, after leaving the heater | 165.48 |

RATE OF COMBUSTION.

| | |
|---|---|
| Pounds of anthracite consumed per hour | 314.343 |
| Pounds of combustible consumed per hour | 271.273 |
| Pounds of anthracite consumed per hour per square foot of grate surface | 8.456 |
| Pounds of anthracite consumed per hour per square foot of water heating surface | 0.365 |
| Pounds of combustible consumed per hour per square foot of grate surface | 7.298 |
| Pounds of combustible consumed per hour per square foot of water heating surface | 0.315 |

VAPORIZATION.

| | |
|---|---|
| Total number of pounds of water that would have been vaporized in the boilers had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the standard atmospheric pressure | 66,052.531 |
| Total number of pounds of water that would have been vaporized in the boilers had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the standard atmospheric pressure | 73,770.158 |
| Number of pounds of water vaporized from 100 degrees by one pound of anthracite | 8.490 |
| Number of pounds of water vaporized from 100 degrees by one pound of combustible | 9.838 |
| Number of pounds of water vaporized from 212 degrees by one pound of anthracite | 9.482 |
| Number of pounds of water vaporized from 212 degrees by one pound of combustible | 10.987 |
| Number of pounds of water vaporized per hour from 212 degrees by one square foot of water heating surface | 3.466 |

There was imparted by the heater to that portion of the feed water which entered it (165.8851—123.4168=) 42.4683 Fahrenheit units of heat, and by the boiler alone to that portion of the feed water which passed out of the heater 1053.8374 Fahrenheit units, making a total of 1096.3057 Fahrenheit units imparted above the temperature of 123.27 degrees Fahrenheit to the portion of the feed water which passed the heater on its way to the boiler. Of this quantity of heat the heater furnished

$(\frac{42.4683 \times 100}{1096.3057} =)$ 3.8738 per centum, and the boiler the remaining 96.1262 per centum.

Now, of the total heat imparted to the feed water in the boiler, above the temperature of 165.48 degrees Fahrenheit for the portion that passed the heater, and above 346.81 degrees for the portion supplied from the steam jackets, namely: $(\overline{61720.4 \times 1053.8374} + \overline{7078.5 \times 875.4080} =)$ 71239841.39096 Fahrenheit units, $(\frac{65043265.86296 \times 100}{71239841.39096} =)$ 91.3018 per centum were imparted to that portion of the feed water which passed through the heater, and 3.8738 per centum of that heat are $(3.8738 \times .913018 =)$ 3.53685 per centum of the whole heat imparted to the whole feed water vaporized in the boiler.

The rate of combustion during the experiment, though extremely limited, was as great as the type of boiler would permit. If the combustion was made more rapid, the boiler primed or foamed to an extent which prevented its economical use.

The deposition of scale upon the lower tube plate (which was also the crown of the furnace) was very troublesome. It was extremely hard, considerable in quantity in a short time, and very difficult of removal.

The merits of the boiler was the small surface of ground it occupied, and the easy means it afforded of superheating the steam. The water circulation was very deficient, and the area for the extrication of the steam very small.

## XXI.

# THE CONTRACT TRIAL OF THE U. S. S. CHARLESTON.

By Passed Assistant Engineer E. T. Warburton, U. S. Navy.

The *Charleston* is a twin screw steel cruiser, with protective deck, of 3,730 tons displacement, built by the Union Iron Works, San Francisco, California.

There are two steel masts, with military tops.

Armament: In a central open superstructure there are six VI-inch B. L. R., three on a side, in projecting sponsons; at the forward and after ends of the superstructure are light barbettes, in each of which are mounted two VI-inch B. L. R. The four barbette guns will later be replaced by two VIII-inch B. L. R. The secondary battery consists of four 6-pounders, two 3-pounders, and two 1-pounder Hotchkiss R. F. guns; four Hotchkiss machine guns and two Gatlings.

There are four above-water torpedo-launching tubes.

The contract required the development for four consecutive hours, by the main and auxiliary engines, of 6,000 I. H. P. for acceptance; subject to a penalty of $100 per I. H. P. below, and a premium of $100 per I. H. P. above 7,000 I. H. P.

The following are some of the principal dimensions and data of the hull:

| | |
|---|---|
| Length, extreme | 320 feet 9¼ inches. |
| Length between perpendiculars | 300 feet. |
| Beam, extreme | 46 feet 2 inches. |
| Depth in hold from top of main deck beams to top of floors | 30 feet 6½ inches. |
| Ratio of length to breadth | 6.49 to 1. |
| Displacement | 3,730 tons. |
| Displacement per inch at L. W. L. | 23 tons. |
| Area L. W. L. | 9,700 square feet. |
| Area of immersed midship section | 741 square feet. |
| Draught of water forward | 17 feet 6 inches. } mean 18 feet 6 inches. |
| Draught of water aft | 19 feet 6 inches. } |
| Coefficient of fineness on midship section | .8677 |

| | |
|---|---|
| Coefficient of fineness on L. W. L. | .66 |
| Coefficient of fineness on extreme dimensions | .51 |
| Transverse meta-centre above centre of gravity | 1.75 feet. |
| Longitudinal meta-centre above L. W. L. (approximate) | 363 feet. |
| Movement to alter trim one inch | 485 foot tons. |

ENGINES.

There are two inclined (7° 21′) direct-acting, compound engines, working at 90 pounds boiler pressure. The engines are duplicates, placed one forward of the other, in separate watertight compartments; the forward engine drives the starboard propeller.

The L. P. cylinder is forward of the H. P. cylinder in the forward compartment and abaft it in the after compartment. The H. P. cylinders and covers are jacketed.

All the valves for the main engines are single-ported piston valves, there being two for each cylinder; the L. P. valves take steam at the middle and exhaust at the ends, and the H. P. valves take steam at the ends and exhaust through the middle to the receivers, which are formed by the cored passages between the H. and L. P. valves.

The valves are operated by the Marshall gear with rockshafts.

The cut-off is varied by altering the travel of the valves by changing the position of a quadrant, which is connected to the reversing engine, and to a single eccentric lever by a radius link.

The reversing gear consists of a horizontal steam cylinder and a hydraulic controlling cylinder, with a floating lever. It can also be operated by hand by means of a small oil pump.

The throttle is a butter-fly valve in the same casing with a double seated equilibrium valve, fitted with relieving valves.

The pistons are solid, of cast steel, in dished shape; the rods are interchangeable, of forged steel.

The crank shafts and pins are made in one forging for each engine, and all the shafting is of forged steel with axial holes through the shafts and crank pins.

The cranks are at 90 degrees with each other, the H. P. crank leading for ahead motion.

The propeller shafts are cased with composition in the bear-

ings in the stern tubes, and they are connected with the thrust shafts by taper ends with sockets in the thrust shafts, and, for each coupling, two feather keys and screwed collar and studs.

The brasses of the shaft bearings and crank pins are lined with Parson's white metal.

The crank pins are lubricated by telescopic oilers in addition to wiper cups.

The pillow blocks of each engine are tied to the cylinders by three rods 6 inches diameter, and two diagonal rods 4 inches diameter, all of forged steel; the crosshead guides also act as ties.

The cast steel crossheads rest on phosphor bronze liners working on cast iron slides, which are cored out for water circulation; the connecting rod thrust in backing is taken by a phosphor bronze box fitted to the crosshead and sliding on the under side of the guide; the sliding faces of liner and box are filled with Parson's white metal in blocks.

There is for each engine a cylindrical sheet copper condenser, with brazed joints; the condensing water circulates through the tubes.

The air pumps are vertical, single acting; there being a double pump for each main engine, worked by a two-cylinder inverted engine directly over the pumps, with crank shaft in bearings above the steam cylinders, and balance turning wheel in the centre.

There are two centrifugal circulating pumps, one in each engine-room, driven by a vertical, inverted, direct-acting engine.

There is also a vertical cast brass auxiliary condenser; it receives some of the auxiliary engine exhausts, and can be put in communication with the starboard main condenser. Its pumps are operated by a two-cylinder inverted engine.

In each engine-room there is a bilge pump with two plungers, driven by a direct-acting steam cylinder controlled by the Dow patent valve gear.

A Dow governor, with floating piston, is fitted to each main engine.

There is a separator in each main steam pipe, and one in the auxiliary steam pipe.

PRINCIPAL DIMENSIONS AND RATIOS OF THE MACHINERY.

| | |
|---|---|
| Number of cylinders (one engine) | 2. |
| Diameter of S.* H. P. cylinder | 44 3/32 inches. |
| Diameter of S. L. P. cylinder | 85 1/16 inches. |
| Diameter of P.† H. P. cylinder | 44 3/32 inches. |
| Diameter of P. L. P. cylinder | 85 1/16 inches. |
| Stroke, common | 3 feet. |
| Diameter of piston rods | 7¼ inches. |
| Diameter of H. P. valves (two for each cylinder) | 13 inches. |
| Area of H. P. steam ports (both valves) for maximum opening | Inboard, 150.4 square inches. Outboard, 149.8 square inches. |
| Area of H. P. exhaust ports (both valves) for maximum opening | Inboard, 219.4 square inches. Outboard, 219.4 square inches. |
| Diameter of L. P. valves (two for each cylinder) | 26 inches. |
| Area of L. P. steam ports (both valves) for maximum opening | Inboard, 348.6 square inches. Outboard, 306.9 square inches. |
| Area of L. P. exhaust ports (both valves) for maximum opening | Inboard, 530.3 square inches. Outboard, 576 square inches. |
| Area of main steam pipe (13½ inches diameter) | 143.14 square inches. |
| Area of main exhaust pipe (24 inches diameter) | 452.39 square inches. |
| Volume swept by S. H. P. piston per stroke | 31.38 cubic feet. |
| Volume swept by S. L. P. piston per stroke | 118.31 cubic feet. |
| Volume swept by P. H. P. piston per stroke | 31.38 cubic feet. |
| Volume swept by P. L. P. piston per stroke | 118.44 cubic feet. |
| Clearance of H. P. cylinder, in per cent. | outboard, 12.5 inboard, 10.75 } mean, 11.62 |
| Clearance of H. P. cylinder, in inches of stroke | outboard, 4.5 inboard, 3.86 } mean, 4.18 |
| Clearance of L. P. cylinder, in per cent. | outboard, 9.2 inboard, 7.72 } mean, 8.46 |
| Clearance of L. P. cylinder, in inches of stroke | outboard, 3.3 inboard, 2.77 } mean, 3.04 |
| Ratio of net area of H. P. piston to L. P. piston | 1 to 3.77 |
| Length of connecting rods between centres | 72 inches. |
| Diameter of connecting rods | 6¼ and 7 inches. |
| Diameter of crank shafts | 12¾ inches. |
| Diameter of crank pins | 12¾ inches. |
| Crank webs, breadth | 15¼ inches. |
| Crank webs, thickness | 8 inches. |
| Diameter of line shafts | 12¼ inches. |
| Diameter of thrust shafts | 12¼ inches. |
| Diameter of propeller shafts | 12¾ inches. |
| Diameter of holes in shafts and crank pins | 4½ inches. |
| Flange couplings, diameter | 23½ inches. |

* Starboard. † Port.

Flange couplings, thickness..... 2¾ inches.
Length of starboard shafting..... 107 feet 5 inches.
Length of port shafting..... 90 feet 2 inches.
Thrust rings, number..... 10
Thrust rings, outside diameter..... 16¾ inches.
Thrust rings, inside diameter..... 12¾ inches.
Thrust surface, in square inches..... 958.
Condenser, length, (total)..... 15 feet 4 inches.
Diameter of shell (outside)..... 5 feet 8 inches.
Thickness of shell..... ¼ inch.
Length of tubes (exposed)..... 13 feet 4½ inches.
Diameter of tubes (outside)..... ⅝ inch.
Number of tubes (one condenser)..... 3,092.
Cooling surface, tubes (one condenser)..... 6,751 square feet.
Auxiliary condenser, cooling surface..... 294.5 square feet.
Air pump engines:
Steam cylinders (2) diameter..... 10 inches.
Air pumps (2) diameter..... 24 inches.
Stroke, common..... 16 inches.
Ratio of volume swept by L. P. piston per stroke to that swept by air pump buckets per stroke..... 14.26 to 1
Circulating pump engine, diameter..... 9 inches.
Circulating pump engine, stroke..... 8 inches.
Diameter of runner..... 32 inches.
Main bilge pumps:
Diameter of steam cylinder..... 9 inches.
Diameter of plungers (2)..... 8¾ inches.
Stroke, common..... 15 inches.

PROPELLERS.

Modified Griffiths—starboard propeller, right hand, and port propeller, left hand; material, manganese bronze; blades, detachable, held in hub by keys in shank and flange of blade fitted under lip of hub with locking pieces; hubs, spherical, with conical tail piece.

Diameter..... 14 feet.
Number of blades (one screw)..... 3.
Pitch for three-fourths of blade:
Entering edge, 16 feet 7 inches.....
Leaving edge, 18 feet 6 inches..... } mean, 17 feet 6 inches.
Tapering to, at root:
Entering edge, 14 feet 6 inches.....
Leaving edge, 16 feet 6 inches..... } mean, 15 feet 6 inches.
Helicoidal area..... 54.75 square feet.
Projected area..... 43.35 square feet.

Disc area..... 153.94 square feet.
Ratio diameter to pitch..... 1 to 1.25
Greatest width of blade at 4 feet radius..... 49⅜ inches.
Diameter of hub..... 42 inches.

BOILERS.

There are in two water-tight compartments six steel, cylindrical, single-ended, horizontal, fire tube boilers with the tubes behind the furnaces leading from a common combustion chamber to a common uptake at the back end. A hanging bridge wall is fitted in the combustion chamber. The uptakes, for each set of three boilers, unite above the water-tight bulkhead between the two boiler rooms, and connect with a single flat-sided standing smoke-pipe.

The air-tight fire rooms are athwartships; there are two blowers in each, drawing air through the ventilators and discharging directly into the fire rooms, without the use of blower casings or air ducts. Each fan is driven by a single cylinder, horizontal engine.

There are two feed pumps of the two-cylinder, double acting piston type, in each fire room—one main and one auxiliary—driven by two-cylinder, inverted engines.

On each boiler there are two Weir's hydrokineters.

There is, in a compartment on the protective deck, one auxiliary steel, cylindrical, single-ended, horizontal return fire-tube boiler, fitted with ash-pan forced draft. The smoke-pipe is inside the main smoke-pipe. There is a No. 2 Davidson pump for feeding the boiler.

Bunkers: The principal coal bunkers are on the protective deck at the sides of the ship; the feeding bunker for each fire room is supplied from the wing bunkers by a railway and a scuttle in the protective deck.

A large part of the space between the wing bunkers and around the boiler hatch is reserved and fitted as auxiliary coal space.

Normal coal supply (mean draft of vessel 18 feet 6 inches)..... 328 tons.
Capacity of feeding bunkers..... 211.8 tons.

Capacity of wing bunkers ..... 370.7 tons.
Capacity of auxiliary bunkers ..... 175.2 tons.
Total coal capacity ..... 757.7 tons.

The following are some of the principal dimensions and ratios of the boilers:

Length ..... 19 feet 3 inches.
Diameter of forward boilers ..... 11 feet.
Diameter of after boilers ..... 11 feet 6 inches.
Number of furnaces in each, (plain, welded, with Adamson ring) ..... 3.
Diameter of furnaces, forward boilers ..... 42½ inches.
Diameter of furnaces, after boilers ..... 44½ inches.
Length of grate, forward boilers ..... 6 feet 1¼ inches.
Length of grate, after boilers ..... 6 feet 1½ inches.
Length of dead plate ..... 5½ inches.
Material of tubes ..... Iron.
Diameter of tubes (outside) ..... 3 inches.
Length of tubes between tube sheets ..... 7 feet 1⅝ inches.
Number of common tubes (one forward boiler) ..... 310.
Number of common tubes (one after boiler) ..... 342.
Thickness of common tubes ..... No. 10 B. W. G.
Number of stay tubes (one forward boiler) ..... 52.
Number of stay tubes (one after boiler) ..... 62.
Thickness of stay tubes ..... $\frac{5}{16}$ inch.
Total number of tubes (one forward boiler) ..... 362.
Total number of tubes (one after boiler) ..... 404.
Depth of combustion chamber ..... 4 feet 6 inches.
Width of combustion chamber (forward boiler) ..... 10 feet.
Width of combustion chamber (after boiler) ..... 10 feet 4 inches.
Height of combustion chamber above upper grate (forward boiler) .. 2 feet 2¾ inches.
Height of combustion chamber above upper grate (after boiler) ..... 2 feet 3¾ inches.
Height of combustion chamber above lower grate (forward boiler) 5 feet 1½ inches.
Height of combustion chamber above lower grate (after boiler) ..... 5 feet 4 inches.
Thickness of shell plates ..... $\frac{11}{16}$ inch.
Thickness of tube sheets ..... $\frac{11}{16}$ inch.
Thickness of front heads ..... { upper plate, ⅞ inch. lower plate, $\frac{13}{16}$ inch.
Thickness of back head above tube sheets ..... ⅞ inch.
Thickness of combustion chamber ..... { top, sides and bottom, $\frac{13}{16}$ inch. front and back ..... $\frac{11}{16}$ inch.
Thickness of furnaces ..... ½ inch.
Riveting: Shell; longitudinal seams, lapped 9⅝ inches and quadruple riveted. Circumferential seams, and seams in heads, double riveted, staggered, with lap joints. All other seams single riveted.
Diameter of rivets for single riveted joints ..... ⅞ inch.

Diameter of rivets for other joints..... $\frac{11}{16}$ inch.
Number of fore and aft braces from head to head..... 18.
Diameter of fore-and-aft braces..... $2\frac{1}{8}$ inches.
Number of fore-and-aft braces, front head to combustion chamber..... 8.
Diameter..... 4 of $1\frac{1}{4}$ and 4 of $1\frac{1}{2}$ inches.
Number of braces from top of combustion chamber to shell..... 44.
Diameter..... $1\frac{1}{2}$ inches.
Load on safety valves, per square inch..... 95 lbs.
Boilers tested to..... 180 lbs.
Tube heating surface, one forward boiler..... 2,027 square feet.
Plate heating surface, one forward boiler..... 377 square feet.
Total heating surface, one forward boiler..... 2,404 square feet.
Tube heating surface, one after boiler..... 2,262.2 square feet.
Plate heating surface, one after boiler..... 382.8 square feet.
Total heating surface, one after boiler..... 2,645 square feet.
Grate surface, one forward boiler..... 68.73 square feet.
Grate surface, one after boiler..... 71.67 square feet.
Area through tubes, one forward boiler..... 14.22 square feet.
Area through tubes, one after boiler..... 15.82 square feet.
Water surface, one forward boiler..... 194 square feet.
Water surface, one after boiler..... 201 square feet.
Steam room, one forward boiler..... 500 cubic feet.
Steam room, one after boiler..... 525 cubic feet.
Capacity of furnaces and combustion chamber, above grates, one forward boiler..... 334 cubic feet.
Capacity of furnaces and combustion chamber, above grates, one after boiler..... 355 cubic feet.
Diameters of smoke-pipe..... 12 feet $5\frac{5}{8}$ inches by 8 feet $11\frac{5}{8}$ inches.
Height of smoke-pipe above lower grate..... 60 feet.

TOTALS FOR SIX MAIN BOILERS.

Heating surface, tube..... 12,867.6 square feet
Heating surface, plate..... 2,279.4 square feet } Total, 15,147 square feet.
Grate surface..... 421.2 square feet.
Area through tubes..... 90.12 square feet.
Area of smoke-pipe (less aux. boiler pipe,)..... 91 square feet.
Area of water surface..... 1,185 square feet.
Capacity of steam room..... 3,075 cubic feet.
Capacity of combustion chambers and furnaces above fire grates..... 2,067 cubic feet.

AUXILIARY BOILER.

Length..... 7 feet 6 inches.
Diameter..... 8 feet.
Number of furnaces (plain, welded, with Adamson ring,)..... 1.
Diameter of furnance..... 3 feet.
Length of grate..... 5 feet.

| | |
|---|---|
| Diameter of tubes (outside,) | 3 inches. |
| Number of common tubes | 68. |
| Thickness of common tubes | No. 10 B. W. G. |
| Number of stay tubes | 16. |
| Thickness of stay tubes | $\frac{1}{16}$ inch. |
| Length of tubes between tube sheets | 5 feet $1\frac{7}{16}$ inch. |
| Load on safety valves | 95 lbs. |
| Boiler tested to | 180 lbs. |
| Tube heating surface | 338 square feet. |
| Plate heating surface | 92 square feet. |
| Total heating surface | 430 square feet. |
| Grate surface | 15 square feet. |
| Area through tubes | 3.26 square feet. |
| Area of water surface | 53 square feet. |
| Capacity of steam room | 92 cubic feet. |
| Capacity of combustion chamber and furnace above grate | 56 cubic feet. |
| Diameter of smoke-pipe | 2 feet. |
| Areas of smoke-pipe | 3.14 square feet. |
| Height of smoke-pipe above grate | 47 feet 6 inches. |

RATIOS OF SIX MAIN BOILERS.

| | |
|---|---|
| Tube H. S. to G. S | 30.55 to 1. |
| Plate H. S. to G. S | 5.41 to 1. |
| Total H. S. to G. S | 35.96 to 1. |
| Area through tubes to G. S | 1 to 4.67. |
| Area of smoke-pipe to G. S | 1 to 4.63. |
| Area of water surface to G. S | 2.8 to 1. |
| Capacity of steam room to G. S | 7.3 to 1. |
| Capacity of combustion chambers and furnaces to G. S | 4.9 to 1. |
| Weight of boilers, fittings and water per 100 square feet of heating surface | 2.198 tons. |

RATIOS OF AUXILIARY BOILER.

| | |
|---|---|
| Tube H. S. to G. S | 22.53 to 1. |
| Plate H. S. to G. S | 6.13 to 1. |
| Total H. S. to G. S | 28.66 to 1. |
| Area through tubes to G. S | 1 to 4.6. |
| Area of smoke-pipe to G. S | 1 to 4.77. |
| Area of water surface to G. S | 3.53 to 1. |
| Capacity of steam room to G. S | 6.13 to 1. |
| Capacity of combustion chamber and furnace to G. S | 3 73 to 1. |
| Weight of boiler, fittings and water per 100 square feet of H. S | 3.42 tons. |

TOTALS FOR SIX MAIN BOILERS AND THE AUXILIARY BOILER, AS USED ON THE TRIAL.

| | |
|---|---|
| Heating surface, tube | 13,205.6 square feet. |
| Heating surface, plate | 2,371.4 square feet. |

| | |
|---|---|
| Heating surface, total | 15,577 square feet. |
| Area of grate surface | 436.2 square feet. |
| Area through tubes | 93.38 square feet. |
| Area of smoke-pipes | 94.14 square feet. |
| Area of water surface | 1,238 square feet. |
| Capacity of steam room | 3,167 cubic feet. |
| Capacity of combustion chambers and furnaces, above grates | 2,123 cubic feet. |

RATIOS OF SIX MAIN BOILERS AND THE AUXILIARY BOILER.

| | |
|---|---|
| Tube H. S. to G. S | 30.27 to 1. |
| Plate H. S. to G. S | 5.43 to 1. |
| Total H. S. to G. S | 35.70 to 1. |
| Area through tubes to G. S | 1 to 4.67. |
| Area of smoke-pipes to G. S | 1 to 4.63. |
| Area of water surface to G. S | 2.83 to 1. |
| Capacity of steam room to G. S | 7.26 to 1. |
| Capacity of combustion chambers and furnaces, above grates, to G. S | 4.86 to 1. |
| Weight of boilers, fittings and water per 100 square feet of H. S | 2.23 tons. |

MAIN (2) AND AUXILIARY (2) FEED PUMPS.

| | |
|---|---|
| Diameter of steam cylinder (2) | 11 inches. |
| Diameter of water cylinder (2) | 6 inches. |
| Stroke, common | 10 inches. |

MAIN BLOWING ENGINES (4).

| | |
|---|---|
| Diameter of cylinder | 9 inches. |
| Stroke | 6 inches. |
| Diameter of fan | 60 inches. |
| Auxiliary boiler feed pump | 4½ x 2½ x 6 inches. |
| Auxiliary boiler blowing engine | 4 x 4 inches. |
| Diameter of fan | 40 inches. |

OTHER AUXILIARY MACHINERY.

In each engine room a two-cylinder engine with two double acting pumps, 6 x 4½ x 7 inches, for bilge, fire and water service and pumping out boilers.

A single, inverted cylinder engine for turning the main engines, 6 x 6 inches.

In each fire room a two-cylinder ash-hoisting engine, 4 x 6 inches.

A two-cylinder, inverted steering engine, working four plunger pumps, with a steam accumulator, 6 x 2¼ x 6 inches.

A two-cylinder, reversible windlass, with two wildcats and two hawser drums, 12 x 10 inches. A two-cylinder reversible winch, with two double rope drums, 9 x 9 inches.

### DISTILLERS AND EVAPORATORS.

Two Baird's double-coil distillers, with filters, and two Baird's evaporators; maximum capacity for twenty-four hours, 6,102.7 gallons.

Circulating pump, also sanitary pump, No. 2 Davidson, 5½ x 6 x 10 inches. Evaporator feed pump, No. 1 Davidson, 3½ x 2 x 4 inches.

### VENTILATING APPARATUS.

2 Sturtevant blowers, 6 x 6 inches, with 54-inch fan.

### ELECTRIC LIGHTS.

There are three Edison compound multipolar marine dynamos of 80 volts, 200 amperes, 400 revolutions, for incandescent and search light plants. Two dynamos are operated by inverted inclined direct-acting compound engines of 40 I. H. P. each; diameter of H. P. cylinder, 7½ inches; diameter of L. P. cylinder, 12 inches; stroke, 6 inches.

The third dynamo is run by an Armington & Sims' double, double-acting horizontal engine, 7 x 5 inches.

There are 439 lamps of 6,400 c. p., of which there are 143 lamps of 1,794 c. p. in the engine department, including bunkers store-rooms, workshop, &c. Four Thomson-Houston projectors will be furnished for search lights.

### WEIGHTS.

| | |
|---|---|
| Propelling machinery | 569.31 tons. |
| Water in above machinery | 116.24 tons. |
| Tools, instruments and duplicate pieces | 24.07 tons. |
| Miscellaneous machinery | 22.50 tons. |
| Total weight at time of completion | 732.12 tons. |
| Weight of one forward boiler, empty, with fittings | 36.35 tons. |
| Weight of water in one forward boiler | 17.63 tons. |
| Weight of one after boiler, empty, with fittings | 38.18 tons. |
| Weight of water in one after boiler | 18.82 tons. |
| Weight of auxiliary boiler, empty, with fittings | 10.49 tons. |
| Weight of water in auxiliary boiler | 4.23 tons. |
| Total weight of all main boilers, with fittings and water | 332.94 tons. |

| | |
|---|---|
| Total weight of all main and auxiliary boilers, with fittings and water... | 347.66 tons. |
| Total weight of all main boilers, with fittings, water and all machinery in fire-rooms..... | 378.21 tons. |
| Total weight of all main boilers and auxiliary boiler, with fittings, water and all machinery in fire-rooms..... | 394.91 tons. |

WEIGHT OF RECIPROCATING PARTS.

| | *Starboard.* | *Port.* |
|---|---|---|
| H. P ..... | 7,479 lbs. | 7,514 lbs. |
| L. P..... | 10,759 lbs. | 11,024 lbs. |

On the first trial trip of the *Charleston*, in May 1889, certain defects in the design of the machinery were developed, and, after a short run in Santa Barbara Channel at full power under forced draft, the trial was abandoned, owing to the heating of the crosshead slides, and the ship returned to the Union Iron Works for the necessary alterations.

The slides were solid forged steel bars, 4½ x 21 inches in section, grooved, at right angles to the line of motion of the crossheads, on top and bottom for oilways; the crosshead liners, or slippers, and the backing boxes were of cast iron. The slides were cut and abraded and particles of cast iron were imbedded in them, and by the action of the heat and applied water, rendered so hard on the surface that it was impossible to properly dress them with the tools at hand.

Another demonstration of the bad character of steel and cast iron when used together as rubbing surfaces, was made on the run to San Francisco, when the port thrust becoming heated, some of the phosphor-bronze rings sprung and the steel collars were forced against the cast iron backing rings with precisely the same result as when the steel slides and cast iron slippers heated.

A few tubes in each condenser leaked, and the air pumps, which were very effective at speeds up to 90 revolutions per minute, making from 26 to 27 inches of vacuum, proved to have insufficient capacity for full speed and the vacuum fell to 22 inches.

The air pump, as shown in the original plans, was a horizontal bucket pump with half trunk, 24 inches diameter and 15 inches stroke, operated together with the bilge pump, by a horizontal compound engine. The pump was made single acting by closing the receiving valves on the outboard end, and the delivery

valves on the inboard end; a change evidently made to overcome a defect in the design. This design was discarded, and the contractors were authorized to make a horizontal, double-acting plunger-pump, 26 inches diameter, retaining the compound engine; but when the pump was first tried at the dock only 21 inches vacuum was made. Various devices were tried to improve the pump, and it was finally changed by substituting a bucket plunger with seven 5¾-inch valves, removing the outboard suction and inboard delivery valves and closing the passages with cast plates and plugs, and filling the clearance space on the outboard end with wood; as already stated, the pump as altered gave excellent results for moderate speeds.

Subsequently to the first trial the following changes were made in the machinery: New cross-head slides of cast iron; slippers and backing boxes of phosphor-bronze, lined with white metal.

White metal blocks let into the backing rings of the thrust bearings.

Double, vertical, single-acting, bucket air pumps, driven by inverted two-cylinder engines.

Independent engines for the main bilge pumps.

New propeller blades, the pitch being reduced one foot throughout. These parts have already been fully described.

On the 21st of August, 1889, the *Charleston*, in charge of the contractors, with the trial boards, and officers detailed as assistants to the Board on Machinery, proceeded to sea for trial.

The contractor's force under Chief Engineer Robert Forsyth was composed of 28 engineers, 17 machinists, 4 water-tenders and oilers, 30 firemen, 4 store-keepers and 1 time clerk, making a total of 84 for the machinery department.

The ship was complete except armament, masts, furniture and stores, and to represent these weights, 470 tons of pig iron was distributed throughout the ship.

The first attempt at trial was interrupted by the overloading of the auxiliary condenser, into which many of the auxiliary engines exhausted. Steam passed through the auxiliary condenser and escaped from the auxiliary feed tank near a main steam pipe joint, and it was at first thought the joint had blown out. An-

other attempt was made the same day, but after a run of over two hours an eccentric of the starboard air pump engine slipped and it was necessary to stop the starboard main engine. The ship was run to San Simeon Bay, remaining there at anchor during the following day. The air pump engine eccentrics were secured; and, it having been found that a few condenser tube glands had worked out, these were replaced and the baffle plates opposite the exhaust openings extended, which prevented further trouble from this source.

On August 23, the machinery was run for six hours at full power, under forced draft, from 1.10 to 7.10 P. M., the last four hours being taken as the period of the official trial.

The auxiliary condenser was in communication with the starboard main condenser, and the auxiliary boiler was connected with the main system.

The machinery and boilers performed perfectly throughout. Water circulated through the cross-head slides and thrust bearings, and, as a precautionary measure, was turned on the crank pins.

The throttle and stop valves were not touched during the trial. Live steam was admitted to the receivers, the valves being one-third open at the start, three-quarters at the end of the first hour, and wide open during the last two hours, and for the latter period the cut-off in the H. P. cylinders was increased from $\frac{7}{10}$ to $\frac{74}{100}$. The boilers steamed freely without priming, and the safety valves were blowing part of the time. All the auxiliary machinery was run continuously without mishap.

Indicator cards were taken every fifteen minutes from the main engines, once an hour from the air pump engines, and at another time the blower, feed pump, circulating pump and main bilge pump engines were indicated after being put in the average condition during the trial. The power of the other auxiliaries was estimated from data taken. The coal burned was Harris' hand-picked navigation (Welsh), and the quantity used was approximated by counting the number of shovelfuls used in an hour in both fire rooms. The speed was measured by a Walker cherub log towed astern.

During the first part of the trial there was a long swell on the

quarter which made the ship roll from moderately to deeply, and caused the engines to race, particularly the starboard engine, the propeller of which was at times partially uncovered. The steering was wild and the ship yawed. The wind was also on the quarter with force 6. After entering Santa Barbara Channel, about 4 o'clock, the sea was smooth, and the force of the wind moderated to 2 to 3, changing in direction to the bow during the latter part of the trial.

After the completion of the trial the ship proceeded to San Pedro for coal, arriving August 24th, and leaving the same day for San Francisco, where she arrived about 8 A. M., August 26th.

On the run up, the speed of the ship being between 12 and 13 knots, the engines making about 77 revolutions per minute, and using the hydraulic steering gear, with the helm hard-a-starboard, from a straight course the direction was changed eight points in 2 minutes 12 seconds, and a circle completed in 6 minutes. With the helm hard-a-port, from a straight course a change in direction of eight points required 2 minutes 27 seconds, and the circle completed in 6½ minutes. To shift the helm from hard-a-port to hard-a-starboard required 36 seconds. The engines were stopped, reversed and ahead again in the following times:

Starboard engine—stop, 20 seconds; back, slow, 28 seconds; full speed astern, 1 minute; stop, 15 seconds; full speed ahead, 1 minute 25 seconds.

Port engine—stop, 30 seconds; back slow, 33 seconds; full speed astern, 1 minute 50 seconds; stop, 25 seconds; ahead slow, 30 seconds; full speed ahead, 1 minute 40 seconds.

DATA OF TRIAL.

| | |
|---|---|
| Draught, mean for trial, forward | 16 feet 7⅔ inches. |
| Draught, mean for trial, aft | 19 feet 1⅓ inches. |
| Draught, mean for trial, mean | 17 feet 10½ inches. |
| Displacement, mean | 3,557 tons. |
| Area immersed midship section, mean | 712 square feet. |
| Wetted surface, Kirk's analysis (at mean draught of 17 feet 10½ inches,) | 17,806.63 square feet. |
| I. H. P. per 100 square feet of wetted surface | 37.43. |
| Distance run in six hours by log | 109 knots. |
| Average | 18.166 knots. |
| Distance run last four hours by log | 72.75 knots. |

MEAN PRESSURES AND I. H. P. FROM

| Card No. | Starboard Engine. | | | | | | | | Revolutions. | Mean pressure in | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Revolutions. | Mean pressure in cylinders. | | | | I. H. P. in cylinders. | | | | H. P. | | |
| | | H. P. | | L. P. | | | | | | | | |
| | | Inboard. | Outboard. | Inboard. | Outboard. | H. P. | L. P. | Total. | | Inboard. | Outboard. | I |
| 9 | 113.7 | 50.42 | 44.71 | 16.01 | 12.46 | 1,480.02 | 1,670.41 | 3,150.43 | 112.9 | 51.67 | 41.27 | |
| 10 | 114.1 | 51.62 | 46.94 | 16.19 | 12.60 | 1,539.04 | 1,695 13 | 3,234.17 | 112.6 | 51.88 | 41.18 | |
| 11 | 112.3 | 47.68 | 42.37 | 15.06 | 11.72 | 1,383.76 | 1,551.92 | 2,935.68 | 111.5 | 49.31 | 40.51 | |
| 12 | 116.2 | 51 82 | 45.53 | 16 34 | 12.71 | 1,547.76 | 1,741 93 | 3,289.69 | 114.8 | 49.73 | 41.18 | |
| 13 | 113.5 | 47.38 | 40.95 | 16.11 | 12 53 | 1,371.57 | 1,677.38 | 3,048.95 | 113.6 | 48.68 | 37.99 | |
| 14 | 115 7 | 48.07 | 41 21 | 15.99 | 12.44 | 1,413.11 | 1,697.41 | 3,110.52 | 114.3 | 50.23 | 40.24 | |
| 15 | 115.6 | 48.59 | 41.12 | 16 56 | 12.89 | 1,418.56 | 1,756.80 | 3,175.36 | 114 4 | 50.14 | 41.86 | |
| 16 | 115.7 | 47.90 | 40.35 | 16.15 | 12.56 | 1,396 64 | 1,714.13 | 3,110.77 | 114.3 | 48.00 | 38.31 | |
| 17 | 116.4 | 49 32 | 41.55 | 16.90 | 13.15 | 1,446.81 | 1,804.99 | 3,251.80 | 114.8 | 47.34 | 37.61 | |
| 18 | 115.4 | 48.85 | 41.55 | 16 43 | 12 78 | 1,427.05 | 1,739.47 | 3,166 52 | 113.6 | 49.65 | 40 56 | |
| 19 | 115. | 48.15 | 43.16 | 15 72 | 12.23 | 1,436.94 | 1,658 65 | 3,095.59 | 114 5 | 49 49 | 39.77 | |
| 20 | 114.3 | 47.79 | 41.26 | 16.36 | 12.73 | 1,392 49 | 1,715.90 | 3,108.39 | 113.9 | 47.29 | 40.69 | |
| 21 | 117.2 | 50.28 | 43 07 | 16.97 | 13.20 | 1,496.68 | 1,824.65 | 3,321 33 | 115.5 | 49.65 | 41.19 | |
| 22 | 117 7 | 46.43 | 38.88 | 17.23 | 13.51 | 1,373.39 | 1,867.08 | 3,240.47 | 114 9 | 48.88 | 41.57 | |
| 23 | 116.6 | 46.63 | 38.67 | 17.06 | 13.01 | 1,360.31 | 1,809.23 | 3,169 54 | 113.9 | 47.90 | 40.55 | |
| 24 | 116.3 | 45.53 | 38.80 | 16.78 | 13.24 | 1,341.63 | 1,801.67 | 3,143.30 | 113.8 | 48 11 | 40.31 | |
| Averages, | 115.35 | 48.53 | 41 88 | 16.37 | 12.73 | 1,426.61 | 1,732.92 | 3,159.53 | 113 95 | 49.25 | 40.30 | |

I. H. P. FROM 3:10 TO 7:10 P. M.

| Port Engine. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Mean pressure in cylinders. | | | | I. H. P. in cylinders. | | | I. H. P. both main engines. | I. H. P. air and circulating pump engines. | I. H. P. other auxiliary engines. | Collective I. H. P. main, air and circulating pump engines. | Collective I. H. P. all machinery in operation. |
| H. P. | | L. P. | | | | | | | | | |
| oard. | Outboard. | Inboard. | Outboard. | H. P. | L. P. | Total. | | | | | |
| 51.67 | 41.27 | 16.72 | 12.19 | 1,434.76 | 1,697 67 | 3,132.43 | 6,282.86 | 114.60 | 213.94 | 6,397.46 | 6,611 40 |
| 51.88 | 41.18 | 16.75 | 11.96 | 1,432.73 | 1,669 79 | 3,102.52 | 6,336.69 | 114.60 | 213 94 | 6,451.29 | 6,665.23 |
| 49.31 | 40.51 | 16 75 | 12.27 | 1,360.65 | 1,671.43 | 3,041.08 | 5,976.76 | 124.15 | 213.94 | 6,100 91 | 6,, 14.85 |
| 49.73 | 41.18 | 16.99 | 12.63 | 1,427.38 | 1,756 30 | 3,183.6[illegible] | 6,473.37 | 124.15 | 213.94 | 6,597.52 | 6,811 40 |
| 48.68 | 37.99 | 16.94 | 12.52 | 1,346.04 | 1,728.73 | 3,074.77 | 6,123.72 | 129.25 | 213.94 | 6,, 52.97 | 6,466 11 |
| 50.23 | 40.24 | 17.63 | 12.94 | 1,413 97 | 1,804 90 | 3,218.87 | 6,329.39 | 129.25 | 213.94 | 6,458 64 | 6,672.58 |
| 50.14 | 41 86 | 17.90 | 12.84 | 1,439 54 | 1,816.45 | 3,255.99 | 6,431.35 | 136.52 | 213.94 | 6,567 87 | 6,781.81 |
| 48.00 | 38.31 | 17.35 | 12.62 | 1,348.92 | 1,769 45 | 3,118.37 | 6,229.14 | 136.52 | 213.94 | 6,365.66 | 6,579.60 |
| 47.34 | 37.61 | 18.39 | 13.45 | 1,333.43 | 1,888.09 | 3,221.52 | 6,473.32 | 143.52 | 213.94 | 6,616 84 | 6,830 78 |
| 49.65 | 40 56 | 17.52 | 13.05 | 1,401.35 | 1,791.89 | 3,195 24 | 6,361.76 | 143.52 | 213.94 | 6,505.28 | 6,719.22 |
| 49 49 | 39.77 | 17.21 | 12.63 | 1,397.53 | 1,767 85 | 3,165.38 | 6,260.97 | 138.73 | 213.94 | 6,399.70 | 6,613.64 |
| 47.29 | 40.69 | 17.08 | 12.64 | 1,370.90 | 1,748.60 | 3,119 50 | 6,227.89 | 138.73 | 213.94 | 6,366.62 | 6,580.56 |
| 49.63 | 41.19 | 17.60 | 12.94 | 1,434 88 | 1,822 06 | 3,256.94 | 6,578 27 | 151.24 | 213.94 | 6,729 51 | 6,943.45 |
| 48.88 | 41.57 | 17.57 | 12.84 | 1,421 66 | 1,804.85 | 3,226.51 | 6,466.98 | 151.24 | 213 94 | 6,618.22 | 6,832.16 |
| 47.90 | 40.55 | 16.99 | 12.38 | 1,378.08 | 1,727.96 | 3,106.04 | 6,275 58 | 151.24 | 213 94 | 6,426.82 | 6,640 76 |
| 48 11 | 40.31 | 16 75 | 12 33 | 1,376.30 | 1,709.33 | 3,085.63 | 6,228.92 | 151.24 | 213.94 | 6,380.16 | 6,594.10 |
| 49.25 | 40.30 | 17.26 | 12.66 | 1,395.45 | 1,,61.08 | 3,156.53 | 6,316.66 | 136.16 | 213.94 | 6,452.22 | 6,6[illegible]0.16 |

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Average ..... 18.187 knots.
Maximum speed by log ..... 18.75 knots.
Average speed for four hours for periods during which indicator cards were taken ..... 18.205 knots.

## Synopsis of steam log of the *Charleston* 23d August, 1889, from 1.10 to 7.10 P. M.

SYNOPSIS OF STEAM LOG OF THE "CHARLESTON," AUGUST 23, 1889, FROM 1:10 TO 7:10 P. M.

(Last columns show averages for official trial.)

| | Six hours. 1:10–7:10 P. M. | | Four hours. 3:10–7:10 P. M. Official trial. | |
|---|---|---|---|---|
| | Forward or starboard. | Aft or port. | Forward or starboard. | Aft or port. |
| Speed for ten-minute periods, during which cards were taken | 18.191 | | 18.205 | |
| Revolutions of main engines | 114.8 | 113.47 | 115.35 | 113.95 |
| Piston speed, feet per minute | 688 8 | 680.82 | 692.10 | 683.70 |
| Steam at boilers (per gauge) | 92.23 | 92.23 | 91.44 | 91.44 |
| Steam at engines (per gauge) | 86.1 | 87.1 | 85.9 | 86.3 |
| Steam at receiver (absolute) | 23.38 | 22.9 | 23.84 | 23.52 |
| Cut-off H. P. cylinder | .712 | .712 | .718 | .718 |
| Cut-off L. P. cylinder | .67 | .665 | .674 | .67 |
| Throttle | Wide. | Wide. | Wide. | Wide. |
| Vacuum in condenser | 26.33 | 26. | 26.17 | 26.06 |
| Temperatures in degrees Fahrenheit — Fire-room | 89.9 | 89.9 | 92.3 | 92.3 |
| Temperatures in degrees Fahrenheit — Engine-room | 104.7 | 88.7 | 107.5 | 90.1 |
| Temperatures in degrees Fahrenheit — Injection | 63.5 | 63.5 | 65.3 | 65.3 |
| Temperatures in degrees Fahrenheit — Discharge | 106.7 | 105. | 107.4 | 106.4 |
| Temperatures in degrees Fahrenheit — Feed | 115. | 112.5 | 116.4 | 115.1 |
| Revolutions of air pump engines | 133.9 | 132. | 134.7 | 135.2 |
| Revolutions of circulating pump engines | 335. | 302. | 343.3 | 313.9 |
| Revolutions of blowers (forward F. R.) | 514. | 482. | 482. | 442. |
| Revolutions of blowers (after F. R.) | 525. | 525. | 510. | 510. |
| Air pressures in fire-rooms in inches of water | 1.62 | 1.82 | 1.57 | 1.98 |

## Average mean pressures, power, &c., for 4-hour trial, 3.10 to 7.10 p. m.

| | | *Starboard.* | *Port.* |
|---|---|---|---|
| H. P. Cylinder | Mean pressure | 45.205 lbs. | 44.775 lbs. |
| | I. H. P. | 1,426.61 | 1,395.45 |
| L. P. Cylinder | Mean pressure | 14.55 lbs. | 14.96 lbs. |
| | I. H. P. | 1,732.92 | 1,761.08 |

| | *Starboard.* | *Port.* |
|---|---|---|
| Aggregate equivalent mean pressure on L. P. piston | 26.54 lbs. | 26.83 lbs. |
| Collective I. H. P. each main engine | 3,159.53 | 3,156.53 |
| Total I. H. P. main engines | 6,316.06 | |
| I. H. P. air pump engines | 21.98 | 20.87 |
| Total I. H. P. air pump engines | 42.85 | |
| I. H. P. circulating pump engines | 51.75 | 41.56 |
| Total I. H. P. circulating pump engines | 93.31 | |
| Collective I. H. P. air and circulating pump engines | 136.16 | |
| I. H. P. main bilge pumps | 6.94 | |
| I. H. P. main feed pumps (4) | 45.84 | |
| I. H. P. main blowers (4) | 75.57 | |
| I. H. P. other auxiliaries | 85.59 | |
| Collective I. H. P. all auxilaries | 350.10 | |
| Collective I. H. P. main engines, air and circulating pump engines | 6,452.22 | |
| Collective I. H. P. main engines and all auxiliary machinery in operation | 6,666.16 | |

Velocity of steam through the ports at the maximum opening of the valves in feet per second:

| | *Inboard.* | *Outboard.* |
|---|---|---|
| H. P. steam ports | 116.5 | 116.99 |
| H. P. exhaust ports | 79.8 | 79.8 |
| L. P. steam ports | 189.6 | 215.4 |
| L. P. exhaust ports | 124.6 | 114.8 |
| Main steam pipe | 122.4 | |
| Main exhaust pipe | 146.1 | |

| | *Starboard.* | *Port.* |
|---|---|---|
| Indicated thrust (I. H. P. main engines,) | 51,651.24 lbs. | 52,236.18 lbs. |
| Indicated thrust per square foot of developed area of propeller (54.75 square feet,) | 94.34 lbs. | 954.08 lbs. |
| Indicated thrust per square inch of thrust rings | 53.9 lbs. | 54.5 lbs. |
| Speed, average for card periods | 18.205 knots. | |
| Slip of propeller | 8 per cent. | |
| $\frac{S^3 \times \text{midship section}}{\text{I. H. P.}}$ | 666.01. | |
| $\frac{S^3 \times D^2}{\text{I. H. P.}}$ | 217.97. | |
| I. H. P., all machinery in operation, per square foot G. S. | 15.28 | |
| I. H. P. per ton of main and auxiliary boilers, water and fittings | 19.17 | |
| I. H. P. per ton of main and auxiliary boilers, water, fittings and all machinery in fire-rooms | 16.88 | |
| I. H. P. per ton of propelling machinery, boilers and water | 9.72 | |
| I. H. P. per ton of total machinery | 9.15 | |
| Square feet of total H. S. per I. H. P. | 2.33 | |

| | *Starboard.* | *Port.* |
|---|---|---|
| Cubic feet swept per I. H. P. per minute by the L. P. pistons... | 8.64 | 8.54 |
| Ratio between the volume swept per minute by the L. P. pistons in double strokes to that of the air pump buckets in single strokes | 24.4 to 1 | 24.05 to 1. |
| Square feet of cooling surface per I. H. P | 2.23 | 2.14 |
| Ratio of cooling surface to total H. S | 1 to 1.129 | |
| Coal, pounds per hour (estimated,) | 20,096. | |
| Coal, pounds per square foot G. S. (estimated,) | 46.07 | |
| Coal, pounds per hour per I. H. P. of main engines, air and circulating pumps (estimated,) | 3.11 | |
| Coal, pounds per hour per total I. H. P. (estimated,) | 3.01 | |

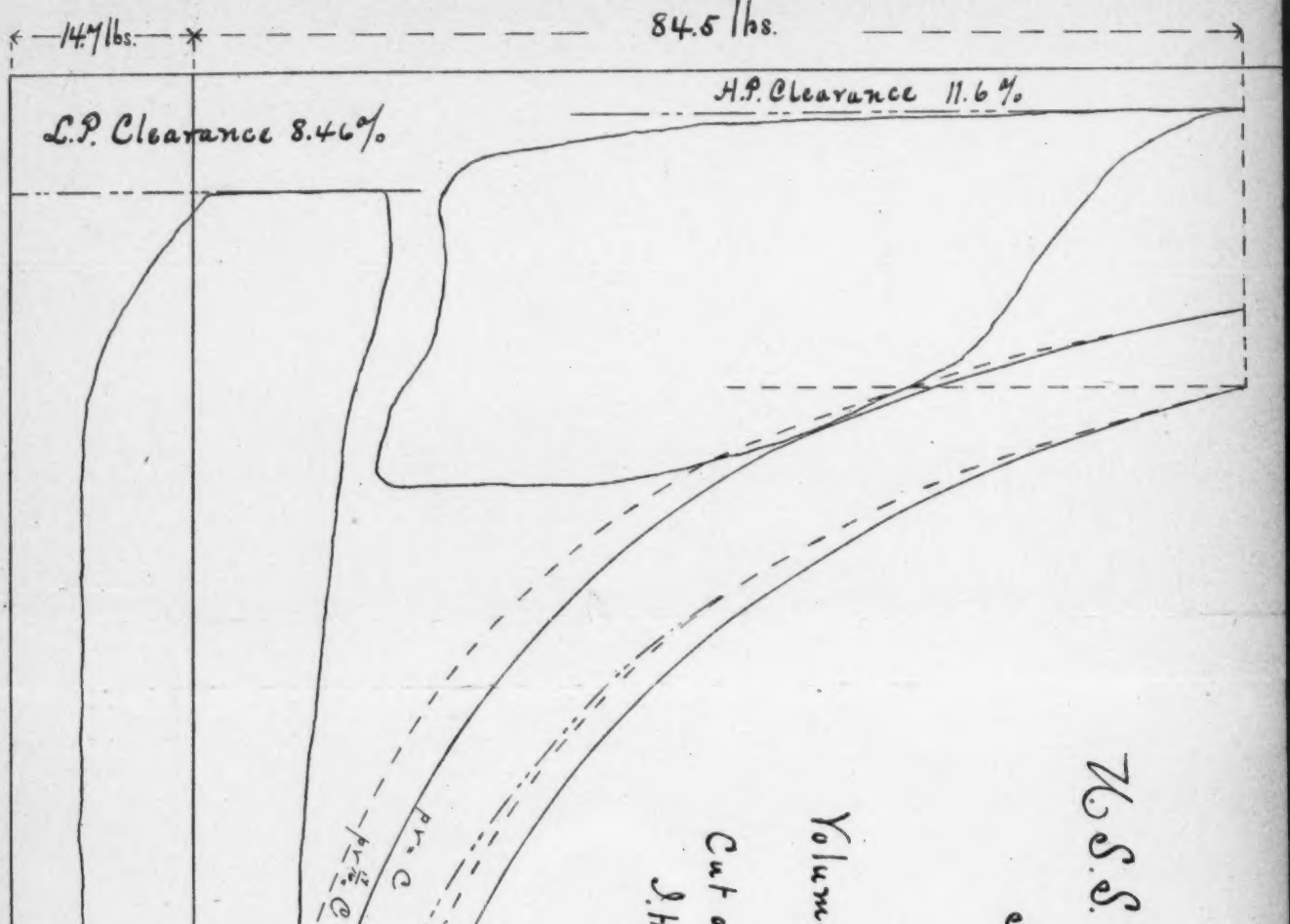
## Maximum—Cards No. 21, 6.10 to 6.20 P. M.

| | *St'b'd or for'd.* | *Port or aft.* |
|---|---|---|
| Revolutions of main engines | 117.2 | 115.5 |
| Piston speed in feet per minute | 703.2 | 693. |
| Steam at boilers (per gauge,) | 90. | 90. |
| Steam at engines (per gauge,) | 85. | 87. |
| Steam at receiver (absolute,) | 23.2 | 22.7 |
| Cut-off H. P. cylinder | .74 | .74 |
| Cut-off L. P. cylinder | .69 | .69 |
| Live steam valve on receivers | Widé open. | |
| Throttle | Wide open. | |
| Vacuum in condensers | 26.25 | 26. |
| Temperatures in degrees Fahrenheit. Fire-room | 92. | 92. |
| Engine-room | 109. | 92. |
| Injection | 68. | 68. |
| Discharge | 110. | 108. |
| Feed | 118. | 118. |
| Revolutions of air-pump engine | 129. | 152. |
| Revolutions of circulating pump engine | 350. | 325. |
| Revolutions of blowers, forward fire-room | 630. | 600. |
| Revolutions of blowers, after fire-room | 480. | 480. |
| Air pressure in fire-rooms, in inches of water | 2¼ | 2. |
| I. H. P., H. P. cylinder | 1,496.68 | 1,434.88 |
| I. H. P., L. P. cylinder | 1,824.65 | 1,822.06 |
| I. H. P., aggregate of each main engine | 3,321.33 | 3,256.94 |
| I. H. P., aggregate of both main engines | 6,578.27 | |
| I. H. P., collective of air and circulating pump engines | 151.24. | |
| I. H. P., blowing engines | 75.57 | |
| I. H. P., feed pump engines | 45.84 | |
| I. H. P., other auxiliaries | 92.53 | |
| I. H. P., collective, main, air and circulating engines | 6,729.51 | |
| I. H. P., collective, main and all auxiliary engines | 6,943.45 | |
| I. H. P., total per square foot of G. S | 15.92 | |
| Square feet of total H. S., per total I. H. P | 2.24 | |

| | *Starboard.* | *Port.* |
|---|---|---|
| Cubic feet swept per I. H. P. per minute by the L. P. pistons | 8.34 | 8.4 |
| I. H. P., total per ton of main and auxiliary boilers, fittings and water | 19.97 | |
| I. H. P., total per ton of main and auxiliary boilers, fittings, water and all machinery in fire-rooms | 17.58 | |
| I. H. P., total per ton of propelling machinery, boilers and water | 10.13 | |
| I. H. P., total per ton of total machinery | 9.48 | |

The *Charleston's* performance for six hours at full power under forced draft, the last hour giving the best results, and the run only being stopped owing to the scarcity of coal and darkness, will compare favorably with the many trials—more than twenty short runs—of her prototype, the *Naniwa-Kan*, before her acceptance by the Japanese Government. From a paper read before an Engineering society by Mr. Marshall, the designer of the machinery, it is seen that the steel slides and cast iron cross-head slippers gave the same trouble that was afterwards experienced on the first trial of the *Charleston*, and that white metal had to be substituted for the rubbing surfaces of the slippers, though on this point no information was conveyed with the drawings purchased by the Navy Department.

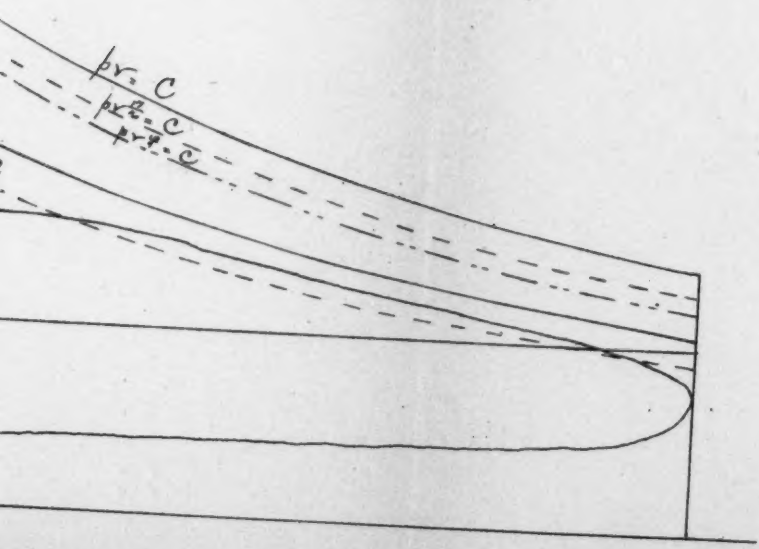
The exact pitch of the propellers used on the trials of the *Naniwa-Kan* is uncertain, and, though the information has been sought, it is not known to the contractors or inspectors of the *Charleston*. It is extremely doubtful, however, judging by the *Charleston's* two trials, whether the mean pitch was 18½ feet as first used on the Charleston, or even 17½ feet, as on the last trial; and this has given rise to the supposition, in the absence of definite information, that the speed and power usually credited to the *Naniwa-Kan* may be due to different propellers; for the reduction in pitch of the *Charleston's* propellers increased the revolutions only eight per minute, and for about 6,650 I. H. P. of the main engines necessary for a total of 7,000 I. H. P. about 120 revolutions will be required, and this number is but little greater than that made by the *Takachiho-Kan* in a trial in Yeddo Bay in May, 1887, when neither the speed nor power of her sister ship, the *Naniwa-Kan*, was developed.

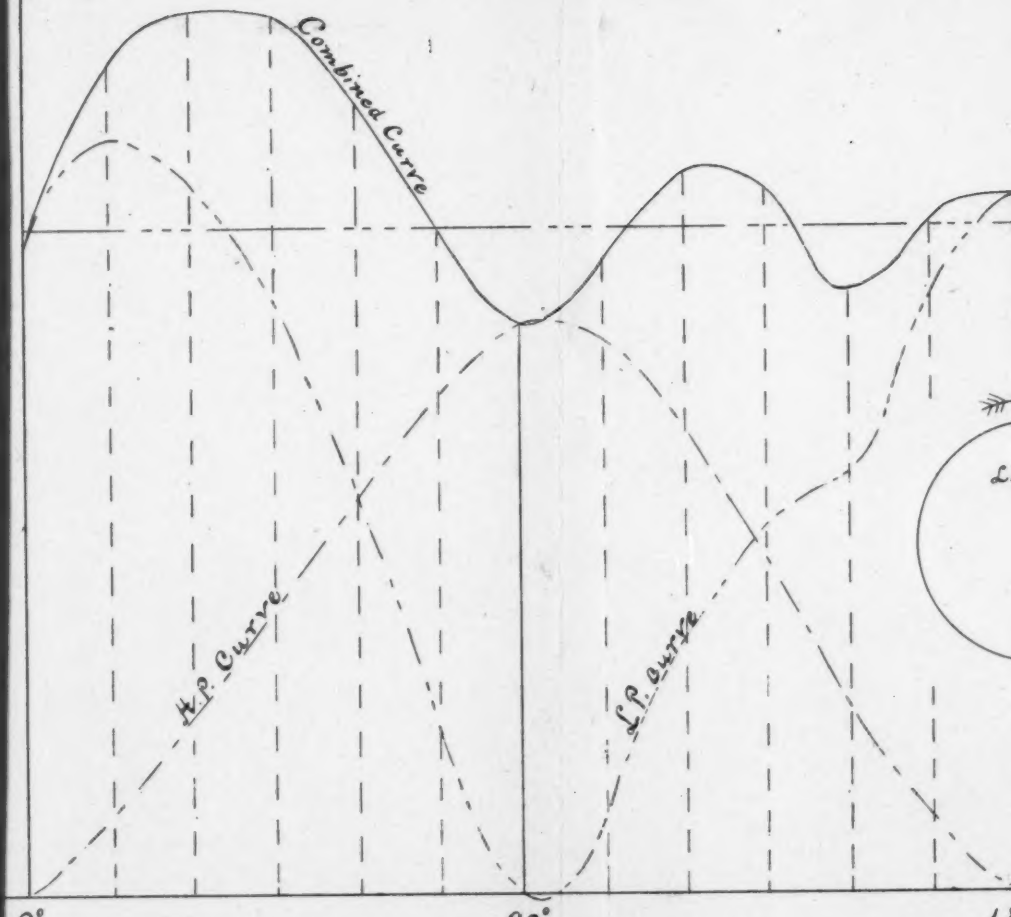

14.7 lbs.
84.5 lbs.
H.P. Clearance 11.6%
L.P. Clearance 8.46%
Volume
Cut
H.P.

S. Charleston

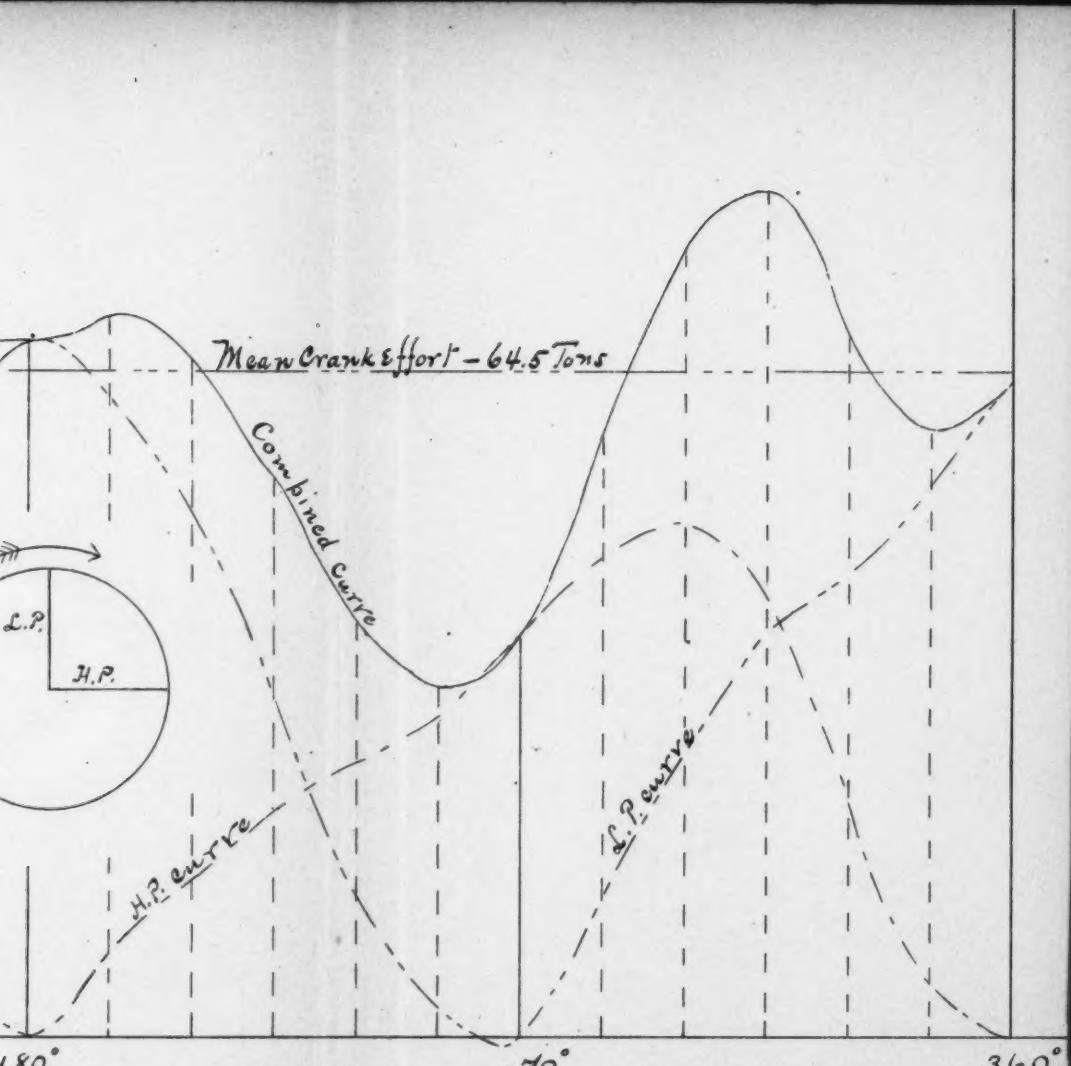
Stbd. Engine - Cards No 22

mes: H.P. cylinder 31.38 cubic feet
L.P. cylinder 118.31 cubic feet
off A.P. cylinder 0.74 stroke
I.H.P. 3240.47






U.S.S. Charleston - Cards No 22
Diagram of Crank Effort. — Str'b'd Engine




I.H.P. = 3240.47. Revs = 117.7

Weight of reciprocating parts

H.P. = 7479 lbs

L.P. = 10759 "

Inclination of engines = 7°-21'

In a translation of an official report of the full speed trial referred to, the synopsis of the steam log gives the following means of four runs over a measured mile, two against and two with the tide:

| | |
|---|---|
| Speed | 17.885 knots. |
| Revolutions, forward engine | 120.5. |
| Revolutions, after engine | 118 |
| Vacuum, forward engine | 24.25 inches. |
| Vacuum, after engine | 24.63 inches. |
| I. H. P. main engines | 5,726 |
| I. H. P. main, air and circulating pump, engines | 5,887.6 |
| I. H. P. all machinery in operation | 5,924.5 |

It is thought that both pitch and surface were reduced by the English builders, and should other screws be made for the *Charleston*, retaining the 17½ feet mean pitch and reducing the diameter one foot, 7,000 I. H. P. could be readily obtained, for the boilers are no doubt capable of furnishing steam for the increased engine speed.

The air pumps also caused much trouble in the *Naniwa-Kan*, and the Union Iron Works received no benefit from the English contractor's experience, nor are the data and description of the pumps in the Japanese vessel definitely known. When the builders of the *Charleston* were allowed to make a radical change in design, they made pumps which were efficient at all speeds, as the synopsis of the steam-log shows, and not only was a high vacuum maintained, but the power of the pump engines was 150 I. H. P. less than that of the compound engines for the horizontal pumps, as developed on the first trial. This amount added to the horse-power made by all the machinery in operation will give 6,816.16 I. H. P. for the mean for four hours, and 7,093.45 I. H. P. for the maximum.

The *Charleston*, on August 21, made 19½ knots by the land, and her record for six hours, as determined by patent log, is high. The true measure of a ship being speed, this excellent quality of the *Charleston* should not be sacrificed for power.

Cards No. 22 were selected for illustration, being next to the maximum, which had a defective card from one end of the starboard L. P. cylinder.

The photographs of drawings of some of the leading features of the machinery, which accompany this account, were made by Passed Assistant Engineer I. N. Hollis, U. S. N., and Mr. W. R. Eckart, members of the Society.

## XXII.

# A COMPRESSED AIR SYSTEM OF FORCED DRAFT.

By Chief Engineer George F. Kutz, U. S. Navy.

[Read at the meeting of the Philadelphia Branch of the Society, Chief Engineer L. W. Robinson, U. S. Navy, in the Chair.]

In 1887 a small steam launch was built at the Mare Island Navy-yard for service in Alaskan waters. The boat was to be cared for by the U. S. S. *Pinta*, and when not in use to be hoisted at the *Pinta's* davits. It was desirable, therefore, as the *Pinta* is a small vessel, to have as little weight as possible in the launch. It was intended that the launch should make trips, as might be necessary, on the coast of Alaska, where, although there is an abundance of fresh water, it is difficult to obtain owing to the rugged nature of the shores. To avoid carrying a large quantity of water in tanks, as well as the weight of the tanks themselves, some system of surface condensation had to be resorted to, and a keel condenser was adopted as being the lightest and least expensive. The propelling engine was compound; cylinders 4 and 7 inches diameter, with a stroke of 6 inches. The boiler was of the "low cylindrical" or direct tubular type, having 2.3 square feet of grate surface. A steam jet in the stack was inadmissible in this case for furnishing the necessary draft, as it would very soon consume the small amount of fresh water carried in the tank. Mr. Klotz, the draughtsman in the Steam Engineering Department of the Yard, suggested the system of using compressed air in a practically open ash pan, as shown by the accompanying plate, and I will here state that I am confident that at that time he had no knowledge of any previous use of compressed air in an open ash pan. Mr. Klotz's arrangement (afterward pat-

ented) consisted in having a small air compressor of the same diameter as the high pressure cylinder, made out of a 4-inch boiler tube, skimmed out smoothly, with composition valve seats screwed on to each end, and the piston driven from the cross-head of the high pressure engine. Hard rubber valves were used in the compressor. The compressed air was conducted through an iron pipe outside of the boiler to the front of the ash pan, where it was delivered into a nozzle, forming part of the ash pan door, through a rectangular slot in the pipe, the pressure being regulated by a wedge valve, by which the amount of opening of the slot could be increased or diminished. Several trials of this launch showed that with a pressure of air of 1½ to 2 pounds per square inch the consumption of bituminous coal (Wellington) of good quality was about 30 pounds per square foot of grate, each trial extending between two and three hours.

It will be understood that the draft by this system of delivering compressed air into a practically open ash pan is very largely an "induced draft." In this particular case the amount of air supplied by the pump or compressor was not more than ten per cent. of that required (theoretically) for the combustion of the fuel, the remaining ninety per cent. being supplied by induction.

The advantages claimed for this system are that the mechanism is inexpensive, with but little liability to derangement. It is economical and noiseless; there is freedom of access to the ash pan for pricking the fire and removal of ashes. As compared with the use of a blower driven by an auxiliary engine, there are points claimed in favor of this system. In the first place, the blower must supply the entire volume of air needed for the combustion of the fuel, the ash pan being closed, that being the usual way by which a blower is used in small launches. The high velocity at which the blower engine must be driven results in rapid wear, and requires no little attention. The first cost of the machinery is considerable, and the amount of oil required for lubrication is not to be neglected; added to these considerations is the humming noise attending the use of the blower, which cannot but be annoying.

It is proper that I should state that this particular boat was

also supplied with a blower—00 monogram—driven direct by a small oscillating engine, and when in use it maintained the steam steadily. The pressure was limited to 100 pounds, and both systems of forced draft furnished all the steam that the engines could use.

Mr. Klotz, as previously stated, secured a patent upon his invention, but he is not alone in this field, for the same system has been applied by a Mr. Anderson, of Glasgow, to vessels of large tonnage running across the Atlantic, one of which is the Cunard Steamship *Servia*. The air compressors used on that vessel are driven by especial compound engines, and there is a reservoir or accumulator, and altogether considerable machinery, necessarily, as the main engines develop some 7,000 I. H. P. I inspected the arrangements on board of the *Servia*, but could obtain no data as to the benefits derived as compared with natural draft. The Engineer stated that he carried steam much more readily, but could give no figures as to relative consumption of fuel.

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## Discussion.

Passed Assistant Engineer J. C. Kafer, U. S. Navy.—The induced draft system by compressed air, described by Chief Engineer Kutz as applied to a steam launch, is a system of supplying air to the fuel at a pressure but little, if any, above the normal amospheric pressure, the ash-pan being open; consequently, the rate of combustion will be but little above that due to natural draft. Where this rate is sufficient, the system described will answer, but I do not think it has any advantage over using the compressed air as a jet in the smoke-pipe instead of steam. A steam jet, such as Korting's, will give sufficient draft to burn 30 pounds of bituminous coal per square foot of grate per hour; and while a steam jet in the smoke-pipe is not desirable, on account of the waste of water involved, the air jet is, and it appears to me that it is much simpler to put the jet in the pipe than to use it below the grate. It would be a very simple and inexpensive experiment to ascertain which of the two methods of using the jet is the more efficient, and I think it would be well worth a trial.

As to the relative economy of this system compared with the usual method of using a blower for supplying all the air, the advantage is with the blower.

The pressure of air required to burn 30 pounds of coal per square foot of grate is not more than one-half inch of water, or $\frac{1}{68}$ of a pound, while the pressure of air used in the jet was from 1½ to 2 pounds, and the quantity supplied at this pressure was estimated at one-tenth of the amount required for combustion.

Assuming that the work done in supplying the air is proportional to the volume of the air multiplied by its pressure, we have, in the Klotz system, one cubic foot × 1.5 = 1.5; and this being asssumed as one-tenth of the amount required for combustion, a blower would have to supply 10 cubic feet at a pressure of .02 of a pound; or 10 × .02 = .2, would represent the relative power for running the blower. In other words, the compressed air jet requires 7½ times as much power for supplying the air necessary for combustion as is required when a blower supplies all the air.

This system can never take the place of the blower, which furnishes all the air required for combustion under a water pressure of two or more inches of water, and where the rate of combustion must exceed 40 pounds of coal per square foot of grate, as is the case with boilers under forced draft in naval steamers; but any system that will get rid of the objectionable features of a closed fire room deserves consideration.

Assistant Engineer F. C. Bieg, U. S. Navy.—In the specifications accompanying his letters patent No. 390,129, September 25, 1888, Mr. Klotz brings in the claim of heating the compressed air during its passage to the nozzle. The jet nozzle is distinct from the induction nozzle, which forms part of the ash-pit door, and surrounds the jet.

Messrs. Anderson and McKinnell use no ash-pit doors. The jet and induction nozzles in the shape of hollow truncated cones are on the pipe which conveys the compressed air, the principal part of the induced air entering the hollow cones. When the ash-pit is to be cleaned or the grates pricked, the pipe holding the nozzles is swung to one side. In Mr. Klotz's arrangement this

can be done without stopping the supply of compressed air. A supplementary pipe and nozzle furnish air to the tops of the fires, but this could be easily added to Mr. Klotz's arrangement.

CHIEF ENGINEER G. F. KUTZ, U. S. NAVY.—In reply to the criticism of Mr. Kafer, I would say that the system described is not intended to supplant the present system of forced draft in use on board naval vessels, but was designed to increase the rate of combustion at small cost for the apparatus employed, with a reasonable degree of economy, and without subjecting the boiler to the destructive working which is employed on many modern naval vessels. I think Mr. Kafer's figures for the relative power required to furnish air under the two systems would hardly stand the test of experiment. The estimate of the amount furnished by the jet as one-tenth of that required for combustion is, of course, a very rough one.

For use on board vessels I would, naturally, recommend an independent cylinder for driving the air compressor, making the system independent of the speed of the main engine, and admitting of a regulation of the draft to suit the requirements of the case. The arrangement described by me, where the compressor is driven from the cross-head of the high-pressure cylinder, was made solely on the score of economy. A test of this system as applied to the boiler in the steam launch of the *Iroquois* was recently made at the Mare Island Yard, and the following particulars of the boat and boiler, and of the compressing apparatus, with the results obtained, will, I am sure, prove interesting; and, as I had no reliable data at the time my paper was read, I take pleasure in submitting the result of this test to show what may be accomplished with the system.

The principal dimensions of the boat are as follows:

| | |
|---|---|
| Length on water line, . . | 28 feet 2½ inches. |
| Breadth, extreme, . . . . | 7 feet 9 inches. |
| Draught, mean during trial, . | 2 feet 7¼ inches. |
| Displacement, . . . . . | 4.6 tons. |

The boiler is a small horizontal, cylindrical, fire-tubular one, with internal furnace, and a 4-inch combustion chamber, beyond which the tubes extend to the back head of the boiler. The

combustion chamber vertically is the same as the diameter of the furnace, but horizontally it is widened so as to permit the use of a large number of tubes. A cylindrical steam drum is riveted to the top at about the middle of the length of the boiler.

The following are the principal data:

| | |
|---|---|
| Working pressure (safe), | 100 pounds. |
| Diameter of shell, | 41 inches. |
| Length, | 36½ inches. |
| Diameter of furnace, | 23½ inches. |
| Length of grate, | 17 inches. |
| Tubes, number, | 161. |
| Outside diameter, | 1½ inches. |
| Length, | 18 inches. |
| Smoke-pipe, diameter, | 8 inches. |
| Height above grate, | 6 feet. |
| Grate surface, | 2.78 square feet. |
| Heating surface, | 90 square feet. |

The compressing cylinder, 4⅛ inches in diameter by 6-inch stroke, is worked from the cross-head of the high pressure cylinder, and projects 3¼ inches forward of the flange of the cylinder. The opening for the admission of air to the ash-pit is 1⅛ inches wide at the narrowest part, and extends the full width of the ash-pit, 11 inches.

While the conditions of weather obtaining when the trial under natural draft was made were not identical with those when the blast was used, the results are such as to leave no question about the efficiency of the apparatus. It must be remembered, in examining the data, that the boiler was designed to burn soft coal with forced draft, and that, therefore, the result is far below what might be accomplished under favorable conditions.

The coal burned was a very hard variety of anthracite, which was slow to ignite. It was carefully weighed during each trial, as was also the water delivered into the hot well by the air pump, the boat being fitted with a keel condenser.

The following are the results obtained:

| | Natural draft. | Induced draft. |
|---|---|---|
| Duration of trial in hours, | 2. | 2. |
| Steam pressure in boiler, | 50. | 60. |
| Vacuum in condenser, | 24. | 24. |
| Revolutions of engines per minute, | 165.1 | 288.5 |
| Temperature of feed water, | 78. | 93. |
| Temperature of water overboard, | 64. | 67. |
| Pounds of coal burned per hour, | 15. | 35. |
| Pounds of water evaporated per hour from the temperature of the feed as measured at the hot well | 88.8 | 275.5 |

While the rate of combustion in both cases was very much slower than it would have been if fairly good soft coal had been used, I think the results are such as to leave no doubt that the system has accomplished what was desired, and that it has been demonstrated that the power of boilers can be largely increased at a trifling expense by the use of this system.

In a private letter to me, Chief Engineer A. S. Greene, U. S. Navy, the senior member of the Board that conducted the trial, says: "With natural draft we were barely able to make three knots, but with the air blast on we easily made six knots, and at one time, after the trial was completed, nearly eight."

As I before remarked, the conditions of weather were not identical, and, therefore, I am not disposed to make any claim on speed.

In my opinion the merit of this system for launches and small craft generally is in its small first cost, its simplicity, noiselessness and efficiency, compared with a blower delivering into a closed ash-pan.

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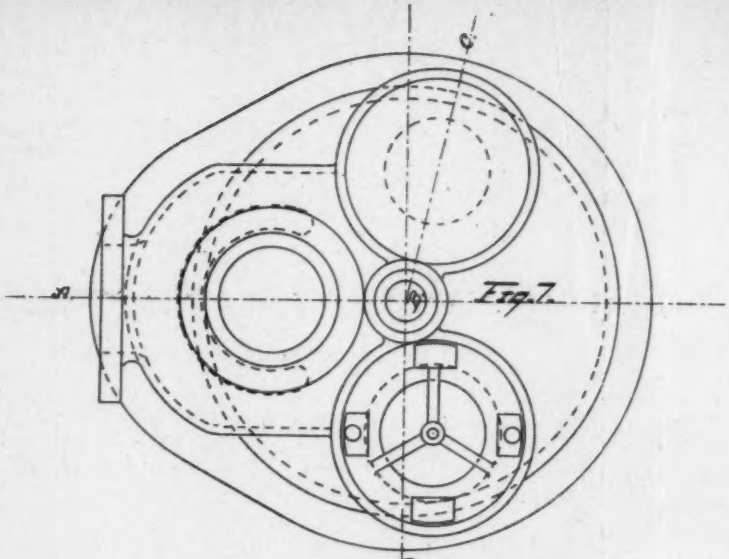
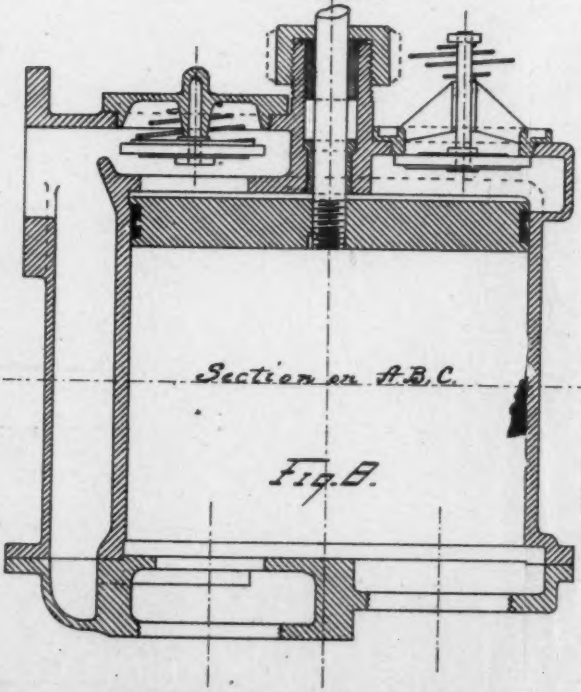



Fig. 7.



Section en A.B.C.

Fig. 8.

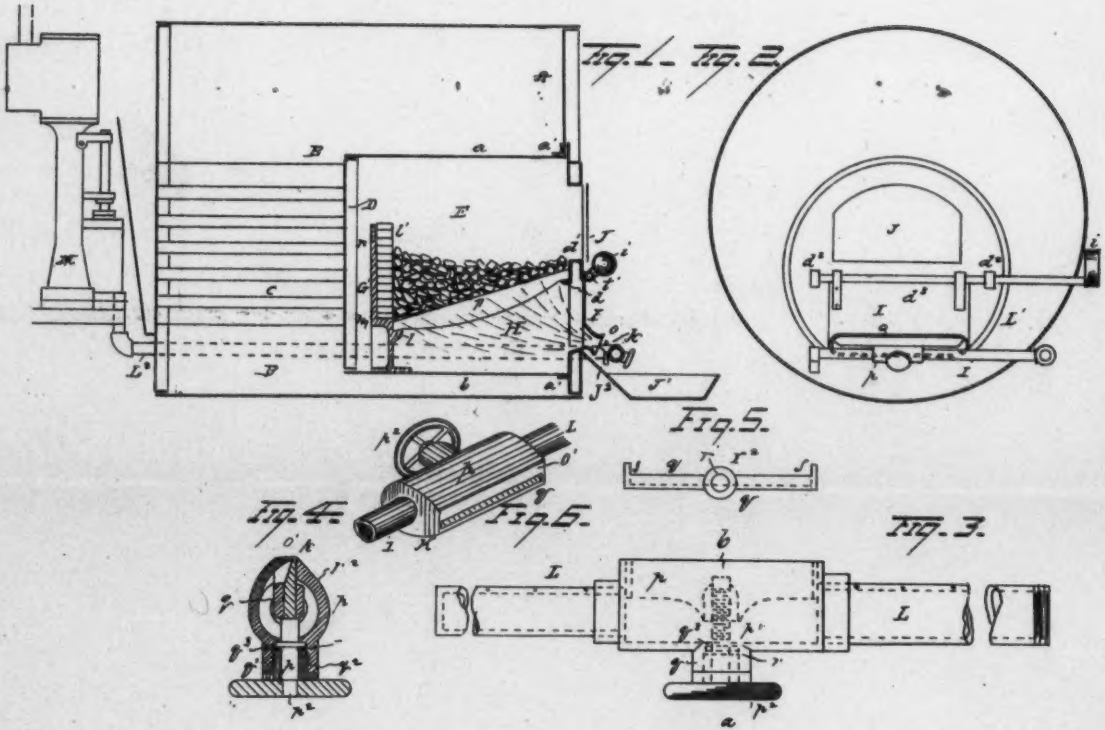


Fig. 1.
Fig. 2.
Fig. 5.
Fig. 4.
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Fig. 3.

## XXIII.

# CONTRACTOR'S FULL POWER FORCED DRAFT TRIAL OF THE U. S. S. "BALTIMORE."

By Assistant Engineers Harold P. Norton and Walter M. McFarland, U. S. Navy.

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The four-hour full power forced draft trial of the *Baltimore's* machinery was made by the contractors at sea, off Cape Henlopen, on September 14, 1889.

The *Baltimore* is a twin-screw protected steel cruiser of 4,400 tons displacement, and was built by the Wm. Cramp & Sons Ship and Engine Building Company of Philadelphia. The plans, known as No. 27 design, were purchased by the Navy Department from Armstrong & Co., of Newcastle-on-Tyne, England. They had been entered in the competition for a swift cruiser for the Spanish Government, which, however, selected those from which the *Reina Regente* was built. The *Baltimore's* machinery was designed by Humphreys & Tennant, but the Messrs. Cramp in the course of building made a number of changes and improvements.

There are two masts fitted with military tops. The main battery consists of two VIII-inch B. L. R. and six VI-inch B. L. R. The secondary battery includes six rapid-fire guns, six revolving cannon and four Gatlings. Five above-water torpedo tubes are fitted.

The following are some of the principal dimensions and data of the hull:

| | |
|---|---|
| Length between perpendiculars | 315 feet. |
| Beam, extreme | 48 feet 6 inches. |
| Area of immersed midship section on trial | 816.64 square feet. |
| Mean draught on trial | 19 feet 6¼ inches. |
| Displacement on trial | 4,391.5 tons. |
| Displacement per inch at L. W. L. | 25.89 tons. |
| Depth in hold from top of main deck beams to top of floors | 31 feet 1.8 inches. |
| Area of load water line plane | 10,875 square feet. |

| | |
|---|---|
| Ratio of length to beam | 6.495 |
| Coefficient of fineness on extreme dimensions | 0.515 |
| Coefficient of fineness on midship section | 0.597 |
| Coefficient of fineness of L. W. L | 0.712 |
| Sail area | None. |
| Transverse metacentre above center of gravity | 2.1 feet. |
| Moment to alter trim one inch | 420 foot tons. |
| Aggregate I. H. P. of main engines and all auxiliaries, as per contract | 9,000. |
| Normal coal supply | 400 tons. |
| Maximum capacity of bunkers | 850 tons. |

*Machinery.*—The design of machinery received from England called for 10,750 I. H. P. for the main engines and all auxiliaries, with the former making 110 revolutions. When the true facts of the numerous trials of the *Naniwa-Kan* were disclosed, the Navy Department decided to reduce the required I. H. P. on a four hours' trial for all the vessels which were about to be contracted for, and that for the *Baltimore* was fixed at 9,000.

There are two horizontal direct-acting triple-expansion engines in separate water-tight compartments; the forward engine drives the port propeller, which is left handed—the starboard one is right handed. The working pressure is 135 pounds.

The main valves are double-ported balanced slides, worked by Stephenson single-bar links. There are no cut-off valves, the variation in expansion being secured by "linking up." In addition, the point of cut-off can be varied for the H. P. cylinder without affecting that of the others, the end of the suspension rod of the H. P. link being articulated to a block adjustable in a slot on the end of the arm on the reversing shaft. All the cylinders are jacketed, reducing-valves being fitted for those of the I. P. and L. P. cylinders.

All shafting, piston rods, and connecting rods are of forged steel, the shafting being hollow. The main pistons are of cast steel, with broad cast iron packing rings. The cross-heads have composition slippers with white metal plugs. The crank shaft and crank pin bearings are fitted with anti-friction metal. The throttle valve is of the butterfly type. There is a starting valve to each clyinder. The reversing gear is actuated by a double-cylinder engine with cranks at right angles. The motion

of this engine is reversed by changing the ports from steam to exhaust and *vice versa*, and there is controlling gear to prevent the link going too far. The engines can also be reversed by hand. Each main crank pin is lubricated by a centrifugal oiler and each cross-head by telescopic gear. The main condensers are of composition, cast in sections and bolted together. The circulating water passes around the tubes. The air pumps are worked directly from the L. P. pistons. The circulating pumps are centrifugal, each driven by a single cylinder direct acting engine. An auxiliary condenser is placed in the forward engine-room, provided with independent air and circulating pumps.

The following are some of the principal dimensions and ratios of the machinery:

| | |
|---|---|
| Number of cylinders (one engine) | 3. |
| Diameter of H. P. cylinder | 42 inches. |
| Diameter of I. P. cylinder | 60 inches. |
| Diameter of L. P. cylinder | 94 inches. |
| Stroke, common | 42 inches. |
| Angle between cranks | 120 degrees. |
| Sequence of cranks (ahead motion) | H. P., I. P., L. P. |
| Area of H. P. steam ports | 144 square inches. |
| Area of H. P. exhaust ports | 176 square inches. |
| Area of I. P. steam ports | 252.5 square inches. |
| Area of I. P. exhaust ports | 366.25 square inches. |
| Area of L. P. steam ports | 487 5 square inches. |
| Area of L. P. exhaust ports | 675. square inches. |
| Area of main steam pipe | 176.7 square inches. |
| Area of main exhaust pipe | 706.9 square inches. |
| Volume swept by H. P. piston per stroke (net) | 32.98 cubic feet. |
| Volume swept by I. P. piston per stroke (net) | 68.03 cubic feet. |
| Volume swept by L. P. piston per stroke (net) | 168. cubic feet. |
| Volumetric clearance in cylinders (about) | 14 per cent. |
| Equivalent of clearance, in inches, of stroke | 5.88 |
| Ratio of net areas of pistons, H. P. to I. P. | 2.063 |
| Ratio of net areas of pistons, I. P. to L. P. | 2.469 |
| Ratio of net areas of pistons, H. P. to L. P. | 5.093 |
| Diameter of piston rods | 8.5 inches. |
| Length of connecting rods between centers | 84 inches. |
| Diameters of connecting rods | 6 and 7 inches. |
| Diameter of crank shaft | 15 inches. |
| Diameter of crank pin | 15½ inches. |

| | |
|---|---|
| Length of crank pin | 16 inches. |
| Diameter of hole in shafts and pins | 7 inches. |
| Width of crank webs | 17 inches. |
| Thickness of crank webs | 11 inches. |
| Diameter of line and thrust shafts | 14 inches. |
| Diameter of hole in above | 8 inches. |
| Diameter of propeller shafting | 15 inches. |
| Diameter of hole in above | 9 inches. |
| Thickness of composition casing of outboard sections of propeller shaft | 1⅛ inches. |
| Diameter of shaft couplings | 29½ inches. |
| Thickness of shaft couplings | 3½ inches. |
| Length of port shafting | 126 feet 5/16-inch. |
| Length of starboard shafting | 97 feet 1⅞ inches. |
| Outside diameter of thrust collars | 19½ inches. |
| Inside diameter of thrust collars | 15¼ inches. |
| Number of bearing thrust collars | 9. |
| Thrust surface | 1,044 square inches. |
| Length of condenser tubes (exposed) | 7 feet 6 inches. |
| Diameter of tubes (outside) | ¾ inch. |
| Number of tubes (one condenser) | 4,256 inches. |
| Cooling surface (one condenser) | 6,184.35 square feet. |
| Cooling surface (auxiliary condenser) | 256.7 square feet. |
| Cooling surface in use on trial | 12,368.7 square feet. |
| Diameter of air pump (double acting) | 28 inches. |
| Ratio of net volume of L. P. cylinder to that of air pump | 11.33 |
| Diameter of centrifugal circulating pump | 42 inches. |

*Screw.*—Type, modified Griffiths, with adjustible blades. Material, manganese bronze.

| | |
|---|---|
| Number of blades | 3. |
| Diameter | 14 feet 6 inches. |
| Pitch, limits of adjustment | 18 feet 6 inches and 21 feet 6 inches. |
| Pitch as set on trial (mean) | 20 feet. |
| Ratio of pitch to diameter | 1.38 |
| Helicoidal area | 57.17 square feet. |
| Projected area | 45 square feet. |
| Disc area | 152.83 square feet. |
| Diameter of hub | 3 feet 11½ inches. |
| Greatest width of blade at a radius of 3 feet 11 inches | 53¼ inches. |

*Boilers.*—There are four double-end cylindrical boilers in two water-tight compartments, there being four furnaces in each end of each boiler. The two furnaces at each end on the same side

of the vertical diameter have a common combustion chamber, making four for each boiler. There are four athwartship fire rooms, which, under forced draft, are supplied with air by two blowers each, driven by Brotherhood engines. The main and the auxiliary feed pumps have double vertical cylinders and a two-throw crank worked by yokes on the piston rods. There are two of each of these pumps. Two oval smoke-pipes are fitted, each 7 feet 7 inches by 5 feet 7 inches. In each boiler compartment there is a small auxiliary boiler.

All the principal parts of the boilers, except the tubes, are of open-hearth steel; the tubes are wrought iron.

The longitudinal seams are treble riveted with double butt straps, and the circumferential, lapped and double riveted; steel rivets are used.

A separator is fitted in each main steam pipe near the engine.

Some of the dimensions of the main boilers are given below:

FOR ONE BOILER.

| | |
|---|---|
| Length | 17 feet 8 inches. |
| Diameter | 14 feet 8 inches. |
| Number of furnaces in each (corrugated) | 8. |
| Diameter of furnaces, greatest internal | 39 inches. |
| Diameter of furnaces, least internal | 36 inches. |
| Length of grate | 6 feet 10 inches. |
| Outside diameter of tubes | 2¼ inches. |
| Length of tubes between sheets | 6 feet 4 inches. |
| Number of ordinary tubes | 712. |
| Number of stay tubes | 236. |
| Total number of tubes | 948. |
| Thickness of ordinary tubes | No. 8, B. W. G. |
| Thickness of stay tubes | ¼ inch. |
| Depth of combustion chamber | 2 feet 1¾ inches. |
| Height of combustion chamber above upper grate | 4 feet 9½ inches. |
| Height of combustion chamber above lower grate | 6 feet 11 inches. |
| Capacity of furnaces and combustion chambers | 487.2 cubic feet. |
| Thickness of shell plates | 1 1/32 inches. |
| Thickness of tube sheets | 11/16 inch. |
| Thickness of front head above tube sheet | ⅜ inch. |
| Thickness of combustion chambers | ½ inch. |
| Thickness of furnaces | ½ inch. |
| Diameter of rivets | 1 1/16 inches. |

| | |
|---|---|
| Number of braces above tubes | 32. |
| Diameter of braces above tubes | 2 inches. |
| Load on safety valves | 135 pounds. |
| Test pressure | 225 pounds. |
| Tube heating surface | 3,597.3 square feet. |
| Plate heating surface | 571.4 square feet. |
| Total heating surface | 4,168.7 square feet. |
| Grate surface | 164 square feet. |
| Area through tubes | 21.68 square feet. |
| Area of water surface | 233.0 square feet. |
| Steam room | 753.0 cubic feet. |
| Weight of one boiler, empty, without grate bars | 54.7 tons. |
| Weight of water in one boiler | 26.3 tons. |
| Weight of water and boiler | 81.0 tons. |
| Area of each smoke-pipe (through armor bars) | 44 square feet. |
| Height of smoke-pipe above lower grates | 61 feet. |

TOTAL FOR THE FOUR MAIN BOILERS.

| | |
|---|---|
| Grate surface | 656 square feet. |
| Heating surface, tubes, 14,389.1 square feet. Heating surface, plates, 2,285.6 square feet. } Total | 16,674.7 square feet. |
| Area through tubes | 86.72 square feet. |
| Capacity of furnaces and combustion chambers above grates | 1,948.8 cubic feet. |
| Steam room | 3,012 cubic feet. |
| Area of water surface | 932 square feet. |
| Weight of boilers, fittings and fire room machinery, including water | 490.66 tons. |
| Weight of propelling machinery, including water | 956.12 tons. |
| Weight of all machinery, tools, spare parts and water, as per specifications | 995.34 tons. |

RATIOS.

| | |
|---|---|
| Tube H. S. to G. S | 21.94 |
| Plate H. S. to G. S | 3.48 |
| Total H. S. to G. S | 25.42 |
| Area through tubes to G. S | 13.22 per cent. |
| Area of smoke-pipes to G. S | 13.41 per cent. |
| Water surface to G. S | 1.42 |
| Steam room per square foot of G. S | 4.59 cubic feet. |
| Capacity of furnaces and combustion chambers per square foot of G. S | 2.97 cubic feet. |
| Weight of boilers and water per 100 square feet of H. S. (in tons) | 2.94 |

AUXILIARY BOILERS.

(These were not used on the trial.)

| | |
|---|---|
| Length | 6 feet 2 inches. |

| | |
|---|---|
| Diameter | 7 feet 2 inches. |
| Number of furnaces in each | 1. |
| Diameter of furnaces | 32 inches. |
| Length of grate | 45 inches. |
| Grate surface, both boilers | 20 square feet. |
| Heating surface, both boilers | 500 square feet. |

*Performance.*—The trial of the *Baltimore* occurred at sea off Cape Henlopen on September 14th, 1889, having been delayed for several days after arrival at the Delaware Breakwater by heavy and foggy weather. The trial began at 8.45 A. M. and continued for four hours, indicator diagrams and observations being taken every fifteen minutes.

The general performance of the machinery was excellent, there being no trouble from heated bearings, although, as a precaution, a small stream of water was kept on the eccentric straps. The boilers steamed freely, but there was a good deal of priming in the forward ones, which occasionally necessitated the partial closing of the throttle of the port (forward) engine. The blowers ran continuously during the four hours. Live steam was not admitted to the receivers at any time.

Pocahontas coal, a free-burning, semi-bituminous coal from the Pocahontas Flat Top Mine, in Tazewell county, Virginia, was used and gave great satisfaction. It was selected lump, clean and of excellent quality, burned freely, made no clinkers, and yielded but a small percentage of refuse. An approximate estimate of the amount of coal used was made, and is given here on account of the almost total lack of any data on the coal consumption of triple-expansion engines, though, of course, it is not so reliable as where the coal has been weighed.

The course of the vessel was to the eastward, with a moderate breeze from the eastward and southward. The sea was moderate, with occasional heavy swells. The speed was registered by a patent taffrail log. As explained further on, an additive correction of three per cent. was obtained, and the speeds given are the readings of the log plus this percentage.

After the completion of the four-hour trial the manœuvering qualities of the vessel were tested, with the following results:

At a speed of 12 knots, with helm hard-a-starboard, completed

circle of 705 yards diameter in 6 minutes 56 seconds. With helm hard-a-port, diameter of circle, 826 yards; time, 7 minutes 4 seconds. With helm hard-a-port, port engine going ahead and starboard engine astern, diameter of circle, 239 yards; time, 7 minutes 8 seconds; revolutions, port, 68; starboard, 64; maximum helm angle in all cases, 40°. When going ahead at 12 knots the ship lost headway in 8 minutes 56 seconds after the signal to stop engines, and sternway was gained in 2 minutes 16 seconds after signal to reverse engines when going ahead 12 knots.

SYNOPOSIS OF STEAM TRIAL OF U. S. S. "BALTIMORE," SEPTEMBER 14, 1889.

(AVERAGES.)

| | | | |
|---|---|---|---|
| Steam at boilers (per gauge) | | 122.2 | |
| | | *Starboard.* | *Port.* |
| Steam at engines (per gauge) | | 117. | 114.4 |
| Steam at 1st receiver (absolute) | | 55.0 | 49.4 |
| Steam at 2d receiver (absolute) | | 18.7 | 19.1 |
| Vacuum in condenser in inches of mercury | | 23.75 | 24.95 |
| Revolutions of main engines per minute | | 118.1 | 118.0 |
| Piston speed, in feet, per minute | | 826.7 | 826.0 |
| Throttle | | Wide. | Wide. |
| Air pressure in fire rooms in inches of water | | 1.66 | 1.66 |
| Revolutions per minute of blowers | | 639. | 639. |
| Temperatures in degrees Fahrenheit. | Fire rooms | 59.2 | 99.9 |
| | Engine rooms | 99.2 | 109.6 |
| | Injection | 67.7 | 67.7 |
| | Discharge | 104.3 | 105.2 |
| | Feed | 122.1 | 130.7 |
| Revolutions of circulating pump | | 182. | 182. |
| Fraction of stroke completed when steam was cut off | H. P. Cylinder | 0.66 | 0.66 |
| | I. P. | 0.77 | 0.77 |
| | L. P. | 0.77 | 0.77 |
| H. P. Cylinder | Mean pressure | 45.86 | 44.73 |
| | I. H. P. | 1,561.84 | 1,520.23 |
| I. P. Cylinder | Mean pressure | 18.25 | 18.12 |
| | I. H. P. | 1,303.52 | 1,273.93 |
| L. P. Cylinder | Mean pressure | 8.61 | 8.78 |
| | I. H. P. | 1,495.92 | 1,522.85 |
| Aggregate equivalent mean pressure on low pressure piston | | 25.19 | 25.08 |
| Collective I. H. P. each mean engine | | 4,361.28 | 4,317.01 |
| Collective I. H. P. both main engines | | 8,678.29 | |
| I. H. P. of circulating pump engines | | 30.16 | 30.16 |

| | *Starboard.* | *Port.* |
|---|---|---|
| I. H. P. of blowers | 152.33 | |
| I. H. P. of feed pumps | 38.92 | |
| I. H. P. of all other auxiliaries | 48.02 | |
| Aggregate I. H. P. of all machinery | 8,977.88 | |
| Cubic feet per minute swept by L. P. piston per I. H. P. | 9.1 | 9.2 |
| Indicated thrust (I. H. P. of main engine only), pounds | 60,940. | 60,365. |
| Indicated thrust per square foot of helicoidal area of propellers (57 17 sq. ft.), pounds | 1,066.0 | 1,056.0 |
| Indicated thrust per square inch of thrust surface, pounds | 58.42 | 57.82 |
| Speed of ship (mean for four hours), knots | 19.575 | |
| Slip of propeller, per cent | 15.96 | 15.94 |
| $\frac{S^3 \times \text{immersed midship section}}{\text{I. H. P.}}$ | 708.05 | |
| $\frac{S^3 \times D^2_3}{\text{I. H. P.}}$ | 232.51 | |
| *Coal used per hour, pounds | 27,383. | |
| *Coal per hour per square foot of G. S., pounds | 41.74 | |
| *Coal per hour per I. H. P. (all machinery), pounds | 3.05 | |
| *Refuse per hour, pounds | 1,643. | |
| *Refuse per hour, per cent | 6.0 | |
| The velocity of the steam through the ports when open wide was, in feet per second— | | |
| H. P. steam port | 129.7 | |
| I. P. steam port | 152.6 | |
| L. P. steam port | 195.2 | |
| H. P. exhaust port | 106.2 | |
| I. P. exhaust port | 105.2 | |
| L. P. exhaust port | 140.9 | |
| Main steam pipe | 105.7 | |
| Main exhaust pipe | 134.2 | |
| I. H. P. per square foot of G. S | 13.68 | |
| Total H. S. ÷ I. H. P | 1.857 | |
| I. H. P. per ton of boilers, water and all fire-room weights | 18.3 | |
| I. H. P. per ton of propelling machinery and water | 9.39 | |
| I. H. P. per ton of all machinery, spare gear, tools and water | 9.02 | |
| Cooling surface per I. H. P | 1.408 | |
| Wetted surface by Kirk's analysis, square feet | 20,302. | |
| I. H. P. per 100 square feet of wetted surface, at 19.575 knots | 42.75 | |
| I. H. P. per 100 square feet of wetted surface, at 10 knots, reduced in ratio of 3.5 power | 4.08 | |

MAXIMUM PERFORMANCE.

| | *Starboard.* | *Port.* |
|---|---|---|
| Steam at boilers (per gauge) | 124.0 | 124.0 |
| Steam at engines (per gauge) | 122.0 | 120.0 |

*Approximate estimate.

| | *Starboard.* | *Port.* |
|---|---|---|
| Steam at 1st receiver (absolute) | 56.7 | 49.77 |
| Steam at 2d receiver (absolute) | 19.7 | 20.27 |
| Vacuum in condensers in inches of mercury | 23.75 | 25.25 |
| Throttle | Wide. | Wide. |
| Revolutions main engines | 119.2 | 122.0 |
| Piston speed in feet per minute | 834.4 | 854.0 |
| I. H. P. of H. P. cylinder | 1,619.87 | 1,627.93 |
| I. H. P. of I. P. cylinder | 1,358.12 | 1,379.77 |
| I. H. P. of L. P. cylinder | 1,560.24 | 1,701.88 |
| Collective I. H. P. each main engine | 4,538.23 | 4,709.58 |
| Collective I. H. P., both main engines | 9,247.81 | |
| I. H. P. circulating pumps | 30.74 | 30.74 |
| I. H. P. blowers | 152.33 | |
| I. H. P. feed pumps | 38 92 | |
| I. H. P. all other auxiliaries | 48.02 | |
| I. H. P. of all machinery | 9.548.56 | |
| Cubic feet per minute swept by L. P. piston per I. H. P | 8.82 | 8.70 |
| Air pressure in fire rooms, in inches of water | 1.8 | |
| Cut off in H. P. cylinder | 0.66 | 0.66 |
| Cut-off in I. P. cylinder | 0.77 | 0.77 |
| Cut-off in L. P. cylinder | 0.77 | 0.77 |
| I. H. P. per square foot of G. S | 14.56 | |
| Total H. S., ÷ I. H. P | 1.746 | |
| I. H. P. per ton of propelling machinery and water | 9.99 | |
| I. H. P. per ton of all machinery, &c | 9.59 | |
| Speed (taken for 15 minutes), in knots per hour | 20.80 | |
| Slip of propeller per cent | 11.58 | 13.61 |
| $\frac{S^3 \times D^3_4}{\text{I. H. P.}}$ | 260.90 | |
| $\frac{S^3 \times \text{immersed midship section}}{\text{I. H. P.}}$ | 794.70 | |

It will be observed at once, on an inspection of the performance coefficients, that they are very high for the mean and remarkably so for the maximum observations. As the horse-power was computed with the utmost care, each card being worked by at least two persons, it becomes interesting to note the method of getting the speed. A Walker's cherub log was used, readings being taken every fifteen minutes. To check these readings observations were made on the way down the Delaware river and bay a few days before the trial, and on the way up afterwards, the runs being at about the same condition of the tide, but in opposite directions with reference to it. By comparing the mean of the distances by log with the actual distance per chart for several

separate distances, it was found that an addition of about three per cent. to the log readings was necessary. These observations were made while the vessel was running at a high speed. During the trial, the log readings were taken by a member of the Board, an officer of high rank, noted for his carefulness in work of this sort. He has kindly explained to the writers in detail all the circumstances, and says, "I believe that, as far as the instruments used would enable us to do so, the speed was determined with perfect fairness, and that, if in error at all, it is understated." There can thus be no question of the speed, except as regards the accuracy of the patent log. On this point there seems to be a consensus of opinion that for accurate indications at high speeds no patent log has yet been constructed on which any dependence can be placed. It is not merely that there is an error, but the error varies from time to time in such an erratic way that it is impossible to correct it by a percentage addition or subtraction. The writers hope, in an early number of the JOURNAL, to discuss this subject, giving some figures of patent log records, which will exemplify what has just been said.

The accompanying table of speeds, revolutions and slips of the propellers based on fifteen-minute observations is added on account of the interest in the data for such high speeds.

It is understood that the Navy Department was entirely willing to accept the vessel as a result of this trial, but the contractors, in the hope of doing better, have asked for and been granted permission to have another trial. Some changes have been made, the most important being the shifting of the propeller blades to their maximum pitch, 21.73 feet for the starboard and 21.4 feet for the port screw. This is done with the idea of making fewer revolutions and carrying a higher boiler pressure so as to get more horse-power. Slight changes about the main valves, to give a freer exhaust have been made, and the air pumps have been overhauled with a view to securing a better vacuum. On this second trial the auxiliary boilers will be used, and they should furnish steam for at least 150 I. H. P. Altogether it would seem as though the contract I. H. P. of 9000 should be readily obtained.

SPEEDS, REVOLUTIONS AND SLIPS OF PROPELLERS.

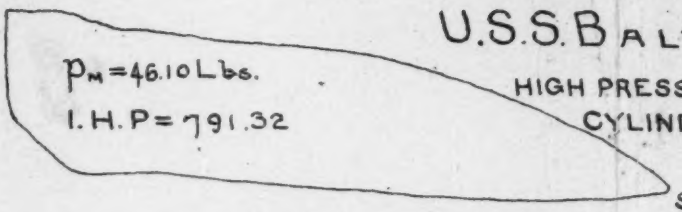
| Beginning of 15-minute period. | Rate of speed per hour in knots during the 15 minute period. | Starboard. | | Port. | |
|---|---|---|---|---|---|
| | | Revolutions of screw per minute. | Slip of screw, per cent. | Revolutions of screw per minute. | Slip of screw, per cent. |
| *A. M.* | | | | | |
| 8.45 | 19.00 | 119.4 | 19.37 | 118.7 | 18.89 |
| 9.00 | 20.00 | 121.0 | 16.25 | 118.6 | 14.55 |
| 9.15 | 19.60 | 120.6 | 17.65 | 115.8 | 14.24 |
| 9.30 | 19.00 | 114.0 | 15.55 | 114.9 | 16.21 |
| 9.45 | 19.00 | 115.6 | 16.72 | 114.7 | 16.07 |
| 10.00 | 19.00 | 120.0 | 19.70 | 117.8 | 18.27 |
| 10.15 | 19.60 | 120.6 | 17.65 | 119.3 | 16.75 |
| 10.30 | 20.00 | 119.0 | 14.84 | 120.7 | 16.04 |
| 10.45 | 20.00 | 119.2 | 14.98 | 120.5 | 15.90 |
| 11.00 | 20.00 | 118.4 | 14.41 | 120.7 | 16.04 |
| 11.15 | 20.00 | 121.2 | 16.39 | 119.9 | 15.48 |
| 11.30 | 20.80 | 119.2 | 11.58 | 122.0 | 13.61 |
| 11.45 | 20.00 | 116.6 | 13.09 | 121.0 | 16.25 |
| 12.00 | 20.00 | 117.8 | 13.97 | 116.3 | 12.86 |
| 12.15 | 18.80 | 113.8 | 16.29 | 112.7 | 15.48 |
| 12.30 | 18.40 | 112.2 | 16.91 | 114.2 | 18.36 |
| Averages | 19.575 | 118.1 | 15.96 | 118.0 | 15.94 |

The accompanying curves of crank effort have been laid out for the port engine when developing the maximum power, but without correcting the cards, the amount of work required for this item being too great for the limited time available. The set of diagrams shown is also uncorrected, but the mean pressures and horse-powers are the correct ones.

STARBOARD

U.S.S. BALT

OUTBOARD.



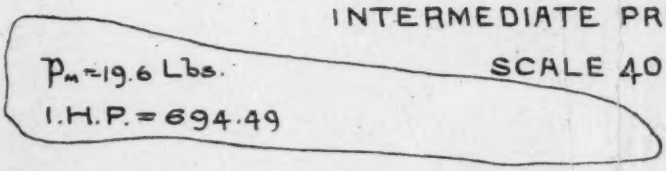

HIGH PRESSU
CYLIND

S

OUTBOARD.

INTERMEDIATE PRES

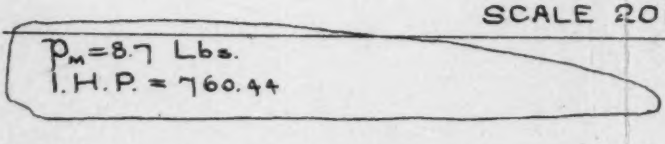
SCALE 40 Lb




OUTBOARD.

LOW PRESSURE C

SCALE 20 Lb



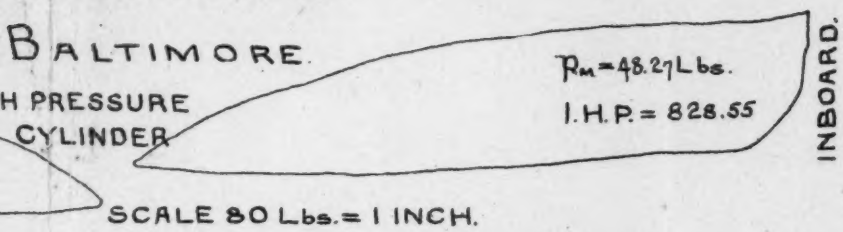

STEAM AT BOILERS 124 Lbs.
STEAM AT ENGINES 122 Lbs.
STEAM AT 1st. RECEIVER 56.7 Lb
STEAM AT 2nd. RECEIVER 19.7 Lb

C

RBOARD ENGINE

BALTIMORE.

H PRESSURE
CYLINDER

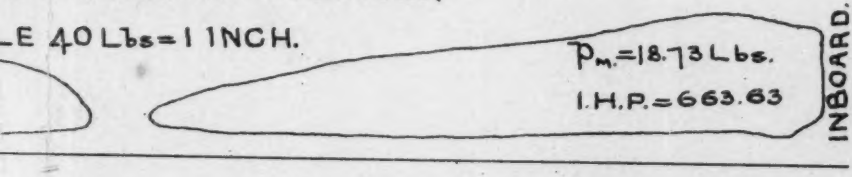



SCALE 80 Lbs. = 1 INCH.

---

TE PRESSURE CYLINDER

LE 40 Lbs = 1 INCH.

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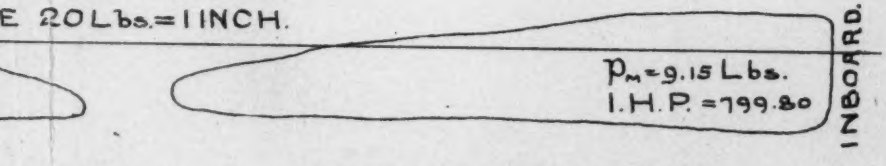
SURE CYLINDER

E 20 Lbs. = 1 INCH.




.4 Lbs.
.2 Lbs.
R 56.7 Lbs. (abs.)
R 19.7 Lbs. (abs.)

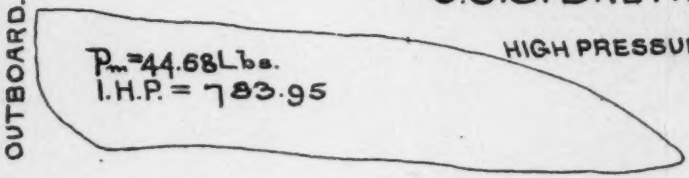
VACUUM 23.75 INCHES.
THROTTLE WIDE OPEN.
REVOLUTIONS 119.2
TOTAL I.H.P. = 4538.23

CARDS Nº 12.

# PORT ENGINE

## U.S.S. BALTIMOR

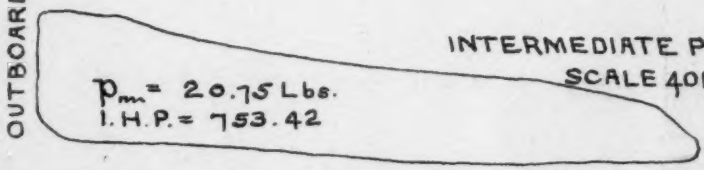
HIGH PRESSURE CY




SCALE 80 Lbs. =

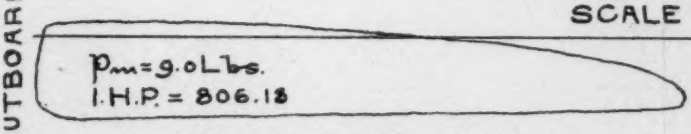
INTERMEDIATE PRES

SCALE 40 Lbs. =




LOW PRESSURE

SCALE 20 L



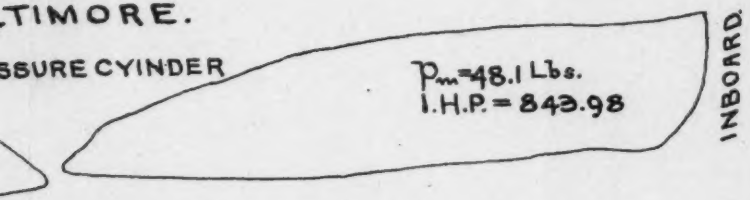

STEAM AT BOILERS 124 Lbs.
STEAM AT ENGINES 120. Lbs.
STEAM AT 1st. RECEIVER 49.77 Lbs. (abs.)
STEAM AT 2nd. RECEIVER 20.27 Lbs. (abs.)

CARDS Nº

.NGINE

.TIMORE.

SSURE CYINDER




0 Lbs. = 1 INCH.

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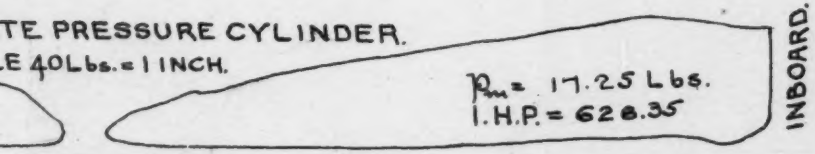
TE PRESSURE CYLINDER.

E 40 Lbs. = 1 INCH.

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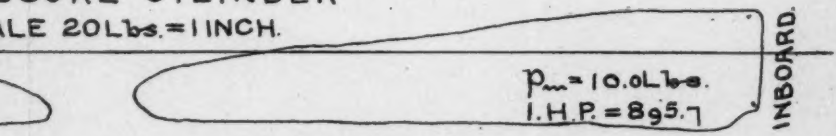
SSURE CYLINDER

ALE 20 Lbs. = 1 INCH.




VACUUM 25 25 INCHES.

THROTTLE WIDE OPEN.

abs.) REVOLUTIONS 122

(abs.) TOTAL I.H.P.= 4709.58

RDS Nº 12.

U.S

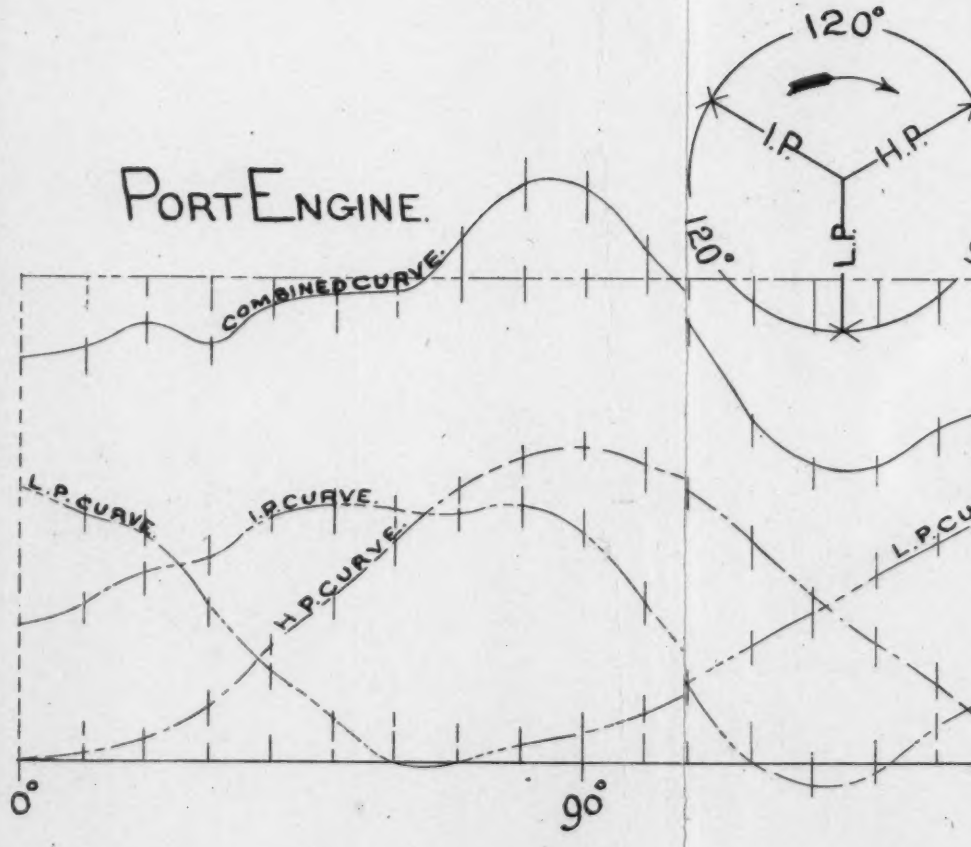
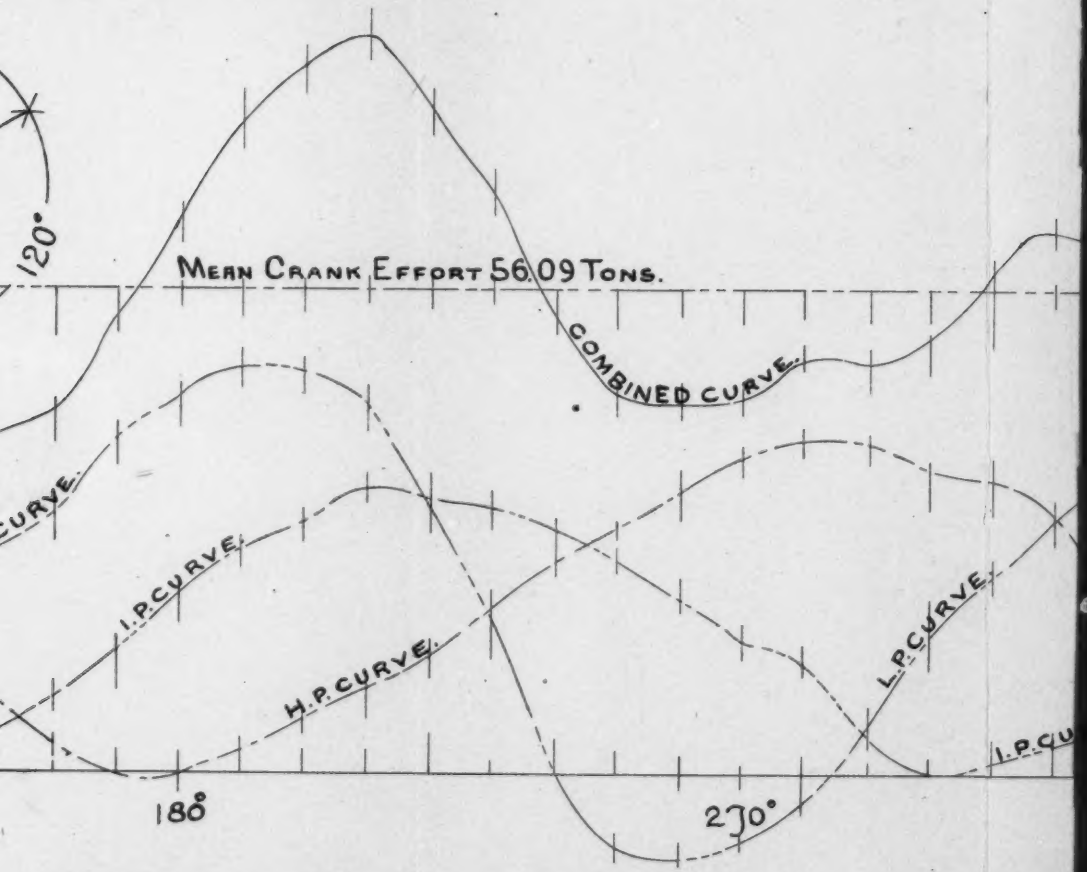



DIAGRAM OF CRANK EFFORT.

MAXIMUN

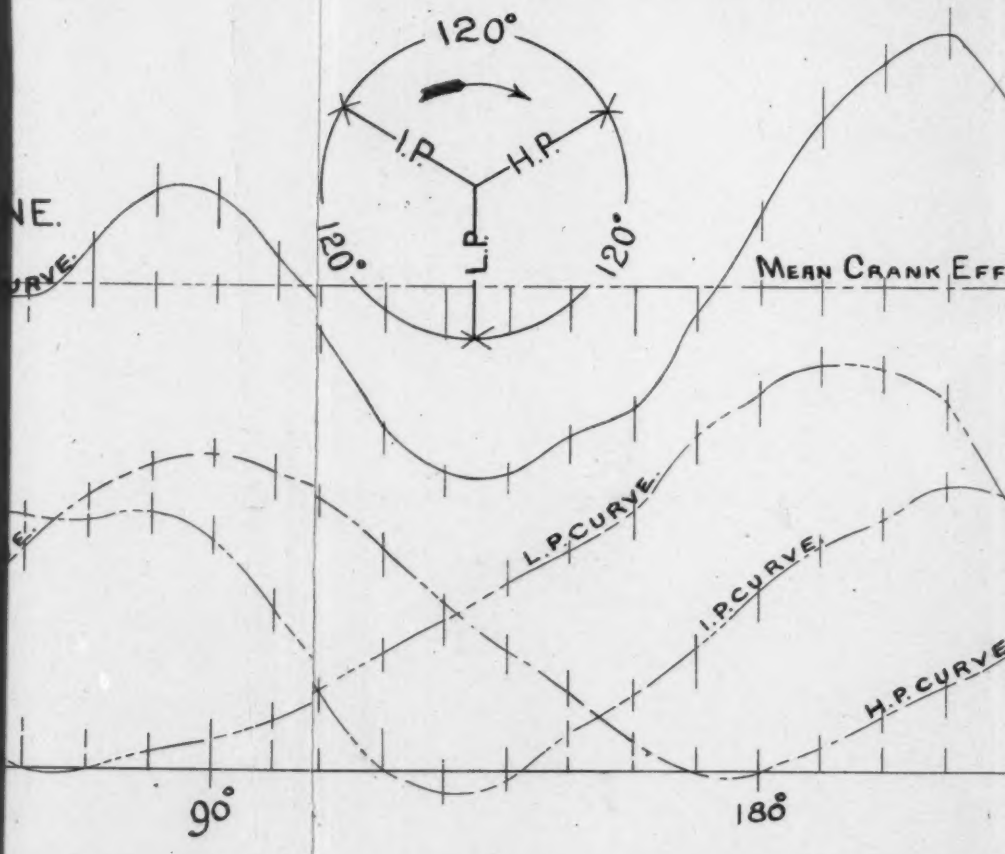
# S.S. Baltimore.




om I.H.P. = 4709.58

Vertical Scale 1 inch = 20

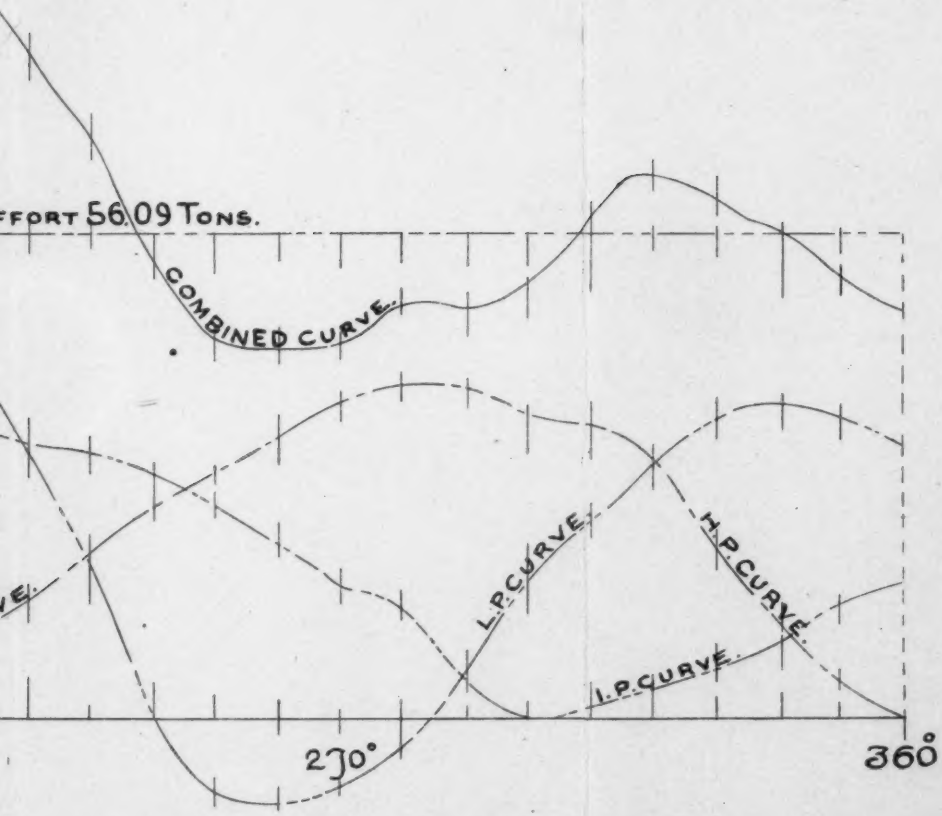
# U.S.S. BALTIMORE




CRANK EFFORT.

MAXIMUM I.H.P. = 4709.58

E.




VERTICAL SCALE 1 INCH = 20 TONS.

# SHIPS.

UNITED STATES.—*3,000-ton Cruisers.*—Two vessels of this displacement were authorized by the act of September 7, 1888. They were advertised June 14, and proposals invited for building them, but when opened on the 22d of August the only bid, that of the William Cramp & Sons' Ship and Engine Building Co., for $1,225,000, being in excess of the amount appropriated, the Department decided to build the vessels and their machinery in the navy-yards. The hull of one (Cruiser No. 7) will be built at the New York navy-yard, and that of the other (Cruiser No. 8) at the Norfolk yard, and the machinery for both at the New York yard.

The following are their principal dimensions:

| | |
|---|---|
| Length, | 300 feet. |
| Beam, | 42 feet. |
| Mean draught, | 18 feet. |
| Displacement at above draught, | 3,183 tons. |
| Tons per inch immersion, | 20 |

They are two-masted schooner rig, spreading 7,210 square feet of canvas, and have poop and forecastle decks connected by bridges along the hammock nettings, with an open gun-deck between.

The protective deck, worked over ½-inch plating, is 2 inches thick on the slope amidships, tapering to 1½ inches at the ends, the thickness on the flat being ½ inch in addition to the ½ inch plate worked all over.

The coal supply at the mean draught of 18 feet is 400 tons, and the bunker capacity 556 tons.

The main battery comprises one 6-inch and ten 4-inch rapid-fire guns on central pivot mounts protected by shields. The 6-inch gun will be placed on the forecastle, two of the 4-inch on

the poop, two under the poop, two under the forecastle and four in broadside, all of the guns on the battery deck being in sponsons. The secondary battery includes two 6-pounder, two 3-pounder, one 1-pounder and two 37 mm. guns. There are six torpedo tubes discharging four feet above water, one forward and one aft being fixed, the others training tubes, all of the Howell pattern.

The engines are twin-screw vertical, triple-expansion, each having four cylinders 36, 53 and two of 57 inches diameter, with a stroke of 33 inches, designed for 10,000 horse-power, with a piston speed of about 900 feet per minute, the H. P. cylinder of each set being placed forward. The crank shafts are in three sections, the two forward sections being interchangeable. The general design of the engines is similar to that for the 2,000-*ton cruisers* described on page 253 of No. 3. The condensers are cylindrical, each having about 7,000 square feet of condensing surface. The air and circulating pumps are independent, the former being vertical and the latter of the centrifugal type.

There are four double-ended and two single-ended boilers, the latter to be used as auxiliaries. Two of the double-ended boilers will be 13 feet 4 inches and two 14 feet 6½ inches diameter, all 20 feet 3½ inches long. The two auxiliary boilers are 11 feet 6 inches diameter and 9 feet 10 inches long. The total grate surface is 616 square feet, and the total heating surface 20,158.

The forced draft is on the closed ash-pit system.

*2,000-ton Cruisers.*—The bids received on the 22d of August for building these vessels being in excess of the amount appropriated ($700,000 each), they were again advertised, the requirements having been modified as follows: Speed reduced from 18 to 17 knots, and a premium of $25,000 offered for each quarter of a knot above 17, and the time increased from two years to two years and six months.

The following were the bids under the terms of the new advertisement:

| | For one. | For two. |
|---|---|---|
| Union Iron Works, San Francisco, | $775,000 | $1,450,000 |
| Bath Iron Works, Bath, Me., | 675,000 | 1,350,000 |
| N. F. Palmer, Jr., & Co., New York, | 674,000 | 1,348,000 |
| Harrison Loring, Boston, | 674,000 | 1,348,000 |
| Columbian Iron Works, Baltimore, | 625,000 | 1,225,000 |

The contracts for two (Cruisers Nos. 9 and 10) have been awarded the Columbian Iron Works, and that for the third (Cruiser No. 11) to Harrison Loring.

*1,000-ton Gunboats* (Nos. 5 and 6).—These vessels are of the following dimensions:

| | |
|---|---|
| Length on water line, | 190 feet. |
| Breadth, extreme, | 32 feet. |
| Mean draught, | 12 feet. |
| Displacement at above draught, | 1,000 tons. |
| Tons per inch immersion, | 9.8 |
| Coal bunker capacity, | 250 tons. |

The engines are twin-screw, vertical triple-expansion, with cylinders 15¾, 22½ and 35 inches diameter by 24 inches stroke, working with a boiler pressure of 160 pounds, and designed for 1,600 I. H. P. with a piston speed of 800 feet per minute. Both sets of engines are in the same compartment and have one condenser in common. There is, in addition, an auxiliary condenser with a capacity equal to all the auxiliary machinery.

There are two boilers of the low cylindrical type, each 9 feet 9 inches diameter and 16 feet 10 inches long, each having three furnaces 36 inches diameter. The total grate surface is 100 and the heating surface 3,630 square feet. The forced draft will be by closed fire-room.

The battery comprises eight 4-inch (33-pdrs.) rapid fire guns; two 47 mm. and two 37 mm. revolving cannon; one rapid fire and one Gatling gun.

The vessels must maintain an average speed of 13 knots per hour for four hours, with an air pressure not greater than one-half inch of water in the fire-rooms. There is a premium of

$5,000 for each quarter of a knot above thirteen, and a like penalty below.

The limit of cost is $350,000 each.

*Practice Vessel.*—This vessel, intended for the use of the cadets at the Naval Academy, is of the following dimensions:

| | |
|---|---|
| Length on water line, | 180 feet. |
| Breadth, extreme, | 32 feet. |
| Draught, mean, | 11 feet 6 inches. |
| Displacement at above draught, | 835 tons. |

She is barkentine rig, spreading 5,000 square feet of canvas.

The engines are similar in all respects to those for the *1,000-ton gunboats*, the cylinders being 13½, 21 and 31 inches diameter by 20 inches stroke, designed for 1,300 I. H. P.

There are two boilers, 8 feet 8 inches diameter and 17 feet long, designed for a working pressure of 160 pounds; each has two 39-inch furnaces. The total grate surface is 78, and the total heating surface 2,654 square feet. A small auxiliary boiler is also fitted. The forced draft will be by closed ash-pits.

The battery consists of four 4-inch rapid fire guns; two 6-pounders, two 3-pounders, and one 1-pounder rapid fire guns, one 37 mm. revolving cannon, and one Gatling.

Under the terms of the advertisement, the vessel must maintain a speed of 12 knots per hour for four consecutive hours, with an air pressure not greater than one-half inch of water in the the ash-pits. The penalties and premiums are the same as for the *1,000-ton gunboats.*

The cost is limited to $260,000.

*Measured Mile Trials at Newport.*—In August and September an elaborate series of speed and turning trials for the cruisers *Chicago, Atlanta, Boston* and *Yorktown* was carried out at Newport.

The measured mile in this place is in many ways most advantageously located. It is completely sheltered, has a minimum depth of 19 fathoms at low water so close to the land that the ranges are established with ease and are readily seen, and there

is ample room with deep water for turning at the ends of the mile. The drawbacks are wind and fog, which frequently delay trials for several days, as was the case with the trials above referred to.

With the exception of the highest speed of the *Atlanta*, all the vessels did well, exceeding in each case their averages on the contract trials. When it is considered that the high speed runs on these trials were practically the first time many of the men had handled modern machinery at full power, the record made is certainly excellent.

Pocahontas coal was used on all these trials and gave entire satisfaction. This coal appears to be growing in favor as it becomes known, and, if future experience continues favorable, it would seem that we have a free-burning semi-bituminous coal as good as the best Welsh.

The performance of the machinery was generally very good. On the *Atlanta* and the *Boston* there was some warming of the bearings, causing a temporary slowing on the former, but not interfering with the latter. On the *Chicago* and *Yorktown* there was no warming whatever. In this respect, the performance of the latter was almost phenomenal. During a run of about an hour at full power and 161 revolutions, there was not a drop of water used on any of the bearings, the adjustment of which was practically perfect. The tradition of "running like a sewing-machine" was exemplified in this case, as the engines ran almost noiselessly.

Some of the more interesting data of the various trials are given below.

*Atlanta.*—

| | | | | |
|---|---|---|---|---|
| Speed, in knots, | 4.45 | 9.89 | 12.77 | 13.45 |
| Revolutions per minute, | 21.06 | 47.82 | 63.82 | 68.80 |
| I. H. P. of main engine, air, circulating and feed pumps, | 160.0 | 904.2 | 2,144.8 | 2,748.5 |
| Indicated thrust, tons, | 4.96 | 12.34 | 21.92 | 26.06 |
| Slip, based on mean pitch, | 5.18 | 7.20 | 10.18 | 12.26 |

During the full power runs the steam pressure in the boilers was 85 lbs; at engines, 72; pressure in receivers (absolute), 34 lbs.; vacuum, 23.5 in.; air pressure in fire-rooms, ¾-inch of water; temperatures—injection 66 degrees, discharge 85 degrees, feed 110 degrees.

*Boston.*—

| | | | | |
|---|---|---|---|---|
| Speed, in knots, . . | 4.61 | 12.00 | 14.35 | 15.58 |
| Revolutions per minute, . | 21.1 | 51.2 | 64.8 | 72.2 |
| I. H. P. of main engine, air, circulating and feed pumps, . . . | 182.95 | 1,178.65 | 2,422.9 | 3,935.9 |
| Indicated thrust, tons, . | 5.24 | 13.95 | 22.66 | 33.04 |
| Slip, based on mean pitch, . | 8.9 | 2.3 | 7.7 | 10.00 |

Averages for four runs at full power: Steam at boilers, 95 lbs.; at engines, 89; receivers (absolute), 33.5; vacuum, 25½; air pressure, 1.7 inches; temperatures—injection 70, discharge 110, feed 120; I. H. P. of blowers and other auxiliaries, 92.5; total I. H. P. of engines and all auxiliaries, 4,028.4.

*Chicago.*—

| Speed in knots. | Revolutions per minute. | | I. H. P. of main engines, air circulating and feed pumps. | | | Indicated thrust in tons. | Slip of screw based on mean pitch. | |
|---|---|---|---|---|---|---|---|---|
| | Stbd. | Port. | Stbd. | Port. | Total. | | Stbd. | Port. |
| 4.33 | 19.56 | 20.36 | 131.05 | 121.50 | 252.55 | 7.61 | 8.77 | 11.86 |
| 10.47 | 47.32 | 46.29 | 826.38 | 614.17 | 1,440.55 | 18 57 | 8.71 | 6.13 |
| 13.27 | 58.85 | 59.86 | 1,461.71 | 1,331.01 | 2,792.72 | 28.31 | 7.01 | 8.04 |
| 15.33 | 69.95 | 70.75 | 2,464.16 | 2,142.05 | 4,606.21 | 39.38 | 9.62 | 10.1 |
| 10.67 | 64.39 | ..... | 2,605.80 | ..... | 2,605.80 | 24.27 | 31.63 | ..... |
| 10.21 | ..... | 63.70 | ..... | 2,095.50 | 2,095.50 | 19.84 | ..... | 33.51 |

Averages for runs at full power:—Steam at boilers, 85, at engines, 80 and 81; receivers (abs.), 40.5 and 39.5; vacuum, 25 and 25; temperatures—injection, 68; discharge, 110; feed, 116.

*Yorktown.*—

| Speed in knots. | Revolutions per minute. | | I. H. P. of main engines, air circulating and feed pumps. | | | Indicated thrust in tons. | Slip of screw based on mean pitch. | |
|---|---|---|---|---|---|---|---|---|
| | Stbd. | Port. | Stbd. | Port. | Total. | | Stbd. | Port. |
| 4.40 | 39.39 | 39.50 | 52.48 | 57.50 | 109.98 | 3.29 | 9.51 | 9.77 |
| 10.61 | 94.16 | 97.75 | 342.41 | 386.20 | 728.61 | 8.95 | 8.61 | 11.98 |
| 14.81 | 140.77 | 140.44 | 1,135.20 | 1,183.73 | 2,318.93 | 18.90 | 14.68 | 14.47 |
| 16.65 | 160.75 | 160.85 | 1,804.20 | 1,774.48 | 3,578.68 | 26.23 | 16.05 | 16.09 |
| 10.96 | 131.42 | ..... | 1,293.45 | ..... | 1,293.45 | 11.60 | 32.36 | ..... |
| 10.63 | ..... | 128.04 | ..... | 1,139.17 | 1,139.17 | 10.49 | ..... | 32.67 |

The I. H. P. of blowers and other auxiliaries at full power was 83.25. I. H. P. of engines and all auxiliaries, 3,661.93.

Averages at full power :—Steam at boilers, 157 ; air-pressure, 2¾ inches forward and 1.9 aft ; steam at engines, 150 and 152 ; first receiver (abs.), 79.8 and 78.8 ; second receiver (abs.), 23.5 and 26.7 ; vacuum, 16.25 and 12.1 ; temperatures—injection, 67 ; discharge, 130; feed, 155.

*Philadelphia.*—This vessel was launched by the William Cramp & Sons Ship and Engine Building Co. on the 7th of September.

*San Francisco.*—Launched by the Union Iron Works on the 26th of October.

*Petrel.*—The contractors' forced draft trial of this vessel took place in Chesapeake Bay on the 19th of August with the following mean results for four hours :

| | |
|---|---|
| Steam pressure in boilers, | 90.0 |
| Steam pressure at engines, | 79.0 |
| Steam pressure in receiver (absolute), | 28.0 |
| Vacuum in condenser, | 23.0 |
| Revolutions, | 126.78 |
| H. P. cylinder, mean pressure, | 52.83 |
| I. H. P., | 527.19 |

| | |
|---|---|
| L. P. cylinder, mean pressure, | 13.98 |
| I. H. P., | 523.38 |
| I. H. P., main engines | 1,050.57 |
| Circulating pump, | 4.27 |
| Blowers, | 35.00 |
| Feed pumps, | 3.57 |
| Other auxiliaries, | 1.74 |
| Total, | 1,095.15 |
| Mean speed, | 11.79 |

The contract power for this vessel was 1,100, and the highest developed during the trial 1,513.

*Nipsic.*—The illustration facing this page represents the stern of this vessel and her screw propeller as they appeared when she was hauled up on the marine railway at Honolulu. The screw was cast at the Washington navy-yard, the mixture being 88 copper, 10 tin and 2 zinc.

This was removed, and a smaller iron screw which had been used on the *Alert*, and which was on hand at the Mare Island navy yard, sent to Honolulu and put on. After completion of repairs, a trial was made with two-thirds boiler power, the engines working up to 62 revolutions per minute, and the mean speed for the trial being 9.2 knots.

*Argus.*—This boat, built for the use of the supervisor of the port of New York, recently had a trial in the Hudson river. She is 85 feet long, 15 feet 9 inches beam, and 9 feet deep, the displacement being 192 tons. She has inverted compound trunk engines, the cylinders being 12 and 21, by 15 inches stroke. The special feature of the boat is the Cowles tubulous boiler, containing 28 square feet of grate, and 820 of heating surface. On the trial, it is said to have furnished an abundance of steam, the speed of the boat reaching 12 knots.

ENGLAND.—A number of the vessels of the *Lapwing* class, described on page 260 of No. 2, have been tried during the past

three months, the *Redpole* being the only one which succeeded on the first trial, nearly all of them having had trouble on the forced draft trial. The results were practically the same as those obtained by the *Lapwing* and *Magpie*, described in No. 3.

The *Basilisk* and *Beagle*, two sheathed gunboats, have recently completed their trials. They are 195 feet long, 30 feet beam, and displace 1,170 tons at a mean draught of 12 feet 5 inches.

The *Basilisk* made 13.1 knots with natural draft on 1,486 I. H. P., and 13.9 with forced draft on 2,107 I. H. P.

The *Beagle* made 13 knots on 1,551, and 13.5 on 2,106 I. H. P.

*Sharpshooter*.—In addition to the trials enumerated on page 262 of No. 3, this vessel made an attempt to run a forced draft trial on the 31st of July; but after 50 minutes, the boilers gave out, this time burning and scalding five men.

Commenting on the numerous failures of this vessel, one of the English service journals says:

"The failure of this vessel on her steam trials, combined with the recent scalding of five men during her last forced draft trial, has caused a considerable amount of disquietude in naval circles. The ultimate utility of this class of vessels is now very much doubted, and as there are several others of exactly the same type, in a more or less advanced stage of completion, some little curiosity is felt concerning the cause of the *Sharpshooter's* long series of mishaps, and the means by which such unsatisfactory results may be avoided in future. Everything connected with those frail vessels is extremely light, and in the machinery department the immense power has been crowded into a very small space, and no further reduction in the weight of the engines could well be made, the Admiralty estimate having been, it is said, cut down. With the excessive vibration set up, due to the high engine speed, abnormal indicated horse-power in proportion to size, slightness of construction, cribbed, cabined and confined boiler space, high steam and air pressure, it is too much to expect her light machinery to pass through such a terribly severe ordeal."

It is now the intention of the Admiralty to make alterations in the machinery of the *Seagull*, a sister vessel, and then subject her to a series of trials with a view to determining to what extent it is prudent to force the engines of this class of vessels.

*Pandora, Pelorus, Persian, Phœnix, Psyche.*—These five cruisers which are intended to form the nucleus of an Australian squadron, are in a forward state of completion. The former three are building by Armstrong, Mitchell & Co., the engines by Hawthorne, Leslie & Co.; and the others, hull and engines, by J. & G. Thomson, Clydebank. The following are the principal dimensions:

| | |
|---|---|
| Length between perpendiculars, | 265 feet. |
| Breadth, extreme, | 41 feet. |
| Draught, mean, | 15½ feet. |
| Displacement, | 2,580 tons. |

They have two masts, without military tops, carry fore and aft steadying sail, and are expected to steam 19 knots per hour on 7,500 horse-power. They have double bottoms, and are fitted with protective decks 2 inches thick on the slope and 1 inch on the flat.

The coal supply is 300 tons.

The engines are twin-screw, vertical, triple-expansion, working with a boiler pressure of 150 pounds, and designed for 4,500 I. H. P. with natural, and 7,500 with forced draft. The cylinders are 30½, 45 and 68 inches diameter by 33 inches stroke. There are four double-ended boilers, each 11 feet 6 inches diameter and 17 feet long, having in all sixteen furnaces with a grate surface of 375 square feet and a heating surface of 9,400 square feet.

The armament will be eight 4.7-inch (45-pounder) rapid-fire guns, eight 3-pounders and several machine guns, besides four torpedo tubes—one on each broadside, one forward, and one aft.

It will be noted that the heating surface of the boilers is very small for the power; and the inevitable result will be, as in some of the recent British vessels, the ruin of the boilers on the contractors' trial in the attempt to get the stipulated power.

There are also two torpedo gunboats of the *Sharpshooter* class, the *Whiting* and the *Wizard*, building for the same squadron by Armstrong & Co. They are of the following dimensions:

Length between perpendiculars, 235 feet; breadth, extreme, 27 feet; draught, mean, 8 feet 3 inches; displacement, 735 tons.

The machinery is by Bellis, and is of the same kind as that of the *Sharpshooter*, described on page 262 of No. 3. The boilers contain 190 square feet of grate and 5,500 square feet of heating surface, and with this it is expected that the I. H. P. will be 4,500, and the speed of the vessel 21 knots. Judging by the performance of the *Sharpshooter*, it will be rather difficult to get the power with these boilers.

The armament will consist of two 4.7-inch and four 3-pounder rapid-fire guns and five torpedo tubes.

The British Admiralty has decided to have three first-class protected cruisers built by contract, one by the Thames Iron Co., another by the Earle Shipbuilding Co., and the third by the Messrs. Napier. Besides these, three similar ones are being constructed in the dock-yards. Their principal dimensions are:

| | |
|---|---|
| Length, . . . . | 360 feet. |
| Beam, . . . . | 60 feet. |
| Draught, mean, . . . | 23 feet 9 inches. |
| Displacement at above draught, | 7,350 tons. |

The engines are of the triple-expansion type, with cylinders 40, 59 and 88 inches diameter by 51 inches stroke, working with a boiler pressure of 155 pounds. The indicated horse-power is expected to be 7,200 with natural and 12,000 with forced draft, and the speed on the measured mile 18 and 20 knots, respectively.

The armament will be two 22-ton, ten 5-ton, twelve 6-pounders and four torpedo tubes.

The protective deck is 5 inches thick over the magazine and machinery.

A correspondent of the *Times* on board the *Northumberland*, writing on the manœuvres, says:

"It is hardly too much to say that this year, as last, the honors of the manœuvres belonged to the *Rattlesnake*. She is never sick nor sorry as a ship of war, and her officers are worthy of her.

It is characteristic of the ways of Whitehall that the contractor who built that ship, which alone of her class has never failed to fulfill the purposes for which she was designed, was actually fined £1,000 sterling for exceeding the weight assigned by her designers to her engines."

FRANCE.—*Forbin.*—This is one of six protected cruisers designed specially for speed, as the dimensions indicate. She is 311⅔ feet long, 30½ beam, draws about 14 feet of water and displaces 1,820 tons. She is schooner rigged, spreading over 7,000 square feet of canvas, and carries two 5½-inch and seven rapid-fire guns and five torpedo tubes. Her coal capacity is only 200 tons. Her engines are twin-screw, horizontal, compound. There are six cylindrical two-furnace boilers, working at a pressure of 100 pounds.

On her recent trials she is reported to have made 20.2 knots as a mean of three runs on the measured mile, and later 20.6 knots on about 5,000 I. H. P.

SPAIN.—A new shipbuilding yard has been started at Bilbao by the Palmer Shipbuilding Co., of England. They are now building three belted cruisers similar to, but larger than those of the *Orlando* class in the English navy, and of the following dimensions: Length, 364 feet; beam, 64; depth, 38; displacement, 7,000 tons. They are expected to make 20 knots on 13,000 I. H. P.

The engines are vertical, twin-screw, triple-expansion, with cylinders 42, 62 and 92 inches diameter by 46 inches stroke. There are six boilers, four of them double-ended, 15 feet 3 inches diameter and 16 feet 3 inches long, and two single-ended, 14 feet 3 inches diameter and 10 feet 6 inches long, all designed for a working pressure of 150 pounds. There are six furnaces in the double and three in each of the single ended boilers, all 42 inches in diameter. The total heating surface is 25,600 square feet, of which 22,000 is in the tubes.

The armor belt is 12 inches thick.

The battery comprises two 28-centimetre, ten 14-centimetre, and eight 6-pounder Hotchkiss guns, and eight torpedo tubes.

## MERCHANT STEAMERS.

*Teutonic.*—The following particulars of this vessel, in addition to those given on pages 167 and 168 of No. 2, will be of interest:

There are sixteen boilers, twelve double and four single-ended; eight of the double-ended boilers are 12 feet 6 inches, and four 12 feet diameter, all 17 feet long; the single-ended boilers are 12 feet 6 inches diameter and 10 feet long. The 12-foot boilers have each four furnaces, the others three furnaces at each end, making a total of 76 furnaces. The diameter of the furnaces in the 12½-foot boilers is 35 inches and the length of grate 61 inches; in the 12-foot boilers, the furnaces are 41 inches diameter and the grates 5 feet long, making a total grate surface of 1,163 square feet. The heating surface is about 39,000 square feet, and the boilers were designed to supply steam at 180 pounds pressure for 17,000 I. H. P.

The original screws were 21 feet 6 inches diameter with a pitch of 28 feet, the helicoidal area of each being 128 square feet. The centers are 16 feet apart athwartships, the blades thus overlapping each other 2 feet 9 inches. With these screws, the revolutions averaged about 70 per minute, but the ship never attained the speed which was expected of her, and during the latter part of September new blades were fitted, by which the diameter was reduced to 19 feet 6 inches and the revolutions increased to 79, resulting in a marked increase of speed. With the new screws she has made three voyages in 6 days 6 hours and 8 minutes, 6 days 6 hours and 27 minutes, and 6 days 7 hours and 3 minutes, respectively, her former time being about 6 days 14 hours.

As a result of the experience with the *Teutonic*, the *Majestic* will be supplied with two more boilers.

The *Columbia*, described on page 168 of No. 2, has nine double-ended boilers, working at a pressure of 150 pounds—

three of them 14 feet 3 inches, and six 15 feet 4 inches diameter, all 17 feet 3 inches long, the smaller boilers having six 38-inch furnaces and the larger ones six 44-inch, the grates of all being about 6 feet 5 inches long. The total grate surface is 1,220 and the total heating surface 35,000 square feet. The main steam pipes, which are of copper, are wrapped with crucible steel wire $\frac{7}{16}$-inch thick.

The cylinders, which are 41, 60 and 101 inches diameter by 66 inches stroke, are steam jacketed. The H. P. and I. P. valves are of the piston type, and the L. P. balanced slides. The crank-shafts are 20½, the line-shaft, 19½, and the propeller shafts 20½ inches diameter. The thrust is of the horse-shoe pattern.

The air-pumps are worked from the H. P. and L. P. engines.

There is an auxiliary condenser in each engine-room furnished with independent air and circulating pumps.

The screws are three-bladed, 18 feet diameter, 32 feet pitch, and each has a helicoidal area of 93 square feet. The diameter of the hub is 54 inches.

On the trial for acceptance, she made 19 knots on 74 revolutions, and is reported to have made 20 on 75½ revolutions. She has maintained an average rate of 18.57 knots per hour for three complete voyages.

La Compagnie Générale Transatlantique has a new steamer, *La Touraine*, in course of construction for its Havre-New York route. She is 516 feet 6 inches long, 55 feet 9 inches beam and 39 feet 8 inches deep, and is to be fitted with triple-expansion, twin-screw engines of 12,500 I. H. P. There are six double and three single-ended boilers in two compartments, one of which contains three of the double and the three single-ended boilers. These boilers are to work with forced draft on the closed ash-pit system. The working pressure is 150 pounds.

Between the two boiler compartments there is a large bunker divided into two water-tight compartments by a tunnel; there is also a bunker forward of the boilers, and one between the after

boilers and the engine-rooms, besides longitudinal bunkers in wake of the boiler spaces, but these are of small capacity.

The Canadian Pacific Railway has made a contract with the Barrow Co. for the construction of three twin-screw steamers to run between Vancouver and Yokohama. They are 440 feet long, 51 feet beam, 36 feet deep, about 7,000 tons displacement, and are intended to make 18 knots per hour.

The same company is about entering into a contract for the construction of three single-screw steamers for their Atlantic route, to run to Halifax in winter, and to Quebec in summer. They will be 480 feet long, 54 feet beam, and will draw 25 feet of water. They are expected to make 20 knots on 10,000 horse-power, and are to be on the route by February 1, 1891.

The Pacific Steam Navigation Company has recently built two steamers, the *Santiago* and the *Arequipa*, for the South American coasting trade. They are double-bottom steamers, 350 feet long, 45 feet beam and 31¾ feet deep, designed for a speed of 14½ knots on 3,500 horse-power. The engines are 31, 49 and 78 by 60 inches stroke, working at a boiler pressure of 160 pounds. There are two double-ended and two single-ended boilers, each 13 feet 9 inches diameter, the former 18 feet 6 inches, and the latter 9 feet 8 inches long.

The *Friesland*, a single-screw steamer building for the Red Star Line by Messrs. J. & G. Thomson, is similar in general features to the *City of Paris*. She is 450 feet long, 51¼ feet beam, 38 feet deep, and of about 6,700 tons. She has a double bottom and is divided into ten water-tight compartments, the only doors in the bulkheads being eight feet above the water line. The engines, which are designed to develop 5,000 I. H. P., are 35½, 56 and 89 by 54 inches stroke, working with a boiler pressure of 160 pounds. There are three double and one single-ended boiler, working under forced draft on the closed stokehold system, besides two donkey boilers for auxiliary machinery.

## CASUALTIES AND REPAIRS.

[It is the purpose of the Council to insert under this head descriptions of ingenious methods of making repairs with the limited facilities available on board ship, and, generally, any matters relating to practical work, or the care of machinery, which seem likely, either from ingenuity or novelty, to be of interest or value.

In sending in sketches illustrating such items, they should be made with black ink on plain white paper, and rolled, not folded.]

PRESERVATION OF SHAFTS.—Contributed by Passed Assistant Engineer Baird.—The outboard portions of the shafts of the twin-screw steamer *Albatross* are naked, except where they pass through the hangers, being covered at these points with short cylinders of hard bronze shrunk on. The bearing is on lignum-vitæ strips dovetailed into short bronze cylinders, which are secured into the wrought-iron hangers. On docking the vessel, after the first season's work, it was found that the paint on the shafts was in good condition except immediately behind the hangers, where corrosion had set in. The shafts were well cleaned and painted, and, when dry, served with marline, over which another coat of paint was applied. At the end of another season the vessel was again docked and the shafts examined, when it was found that the corrosion had continued. The corroded portions were again cleaned and painted, and then covered with the cotton tape used for insulating electric wires. This tape had been steeped in a black "insulation compound" made by the Edison Light Company, and was put on in two layers, the edges lapped, and each layer breaking joints with the other. The tape was then served with fishing twine and painted, the length covered being about 18 inches. After another season's work the shaft under the tape was found to be entirely free from corrosion. Portions of the shafts under the tape were thereafter examined every time the vessel was docked, and always with the same result.

A PROBABLE CASE OF SLOW COMBUSTION OF IRON.—Contributed by Chief Engineer McKean.—Some years ago one of our

vessels was stationed for the winter in a severe climate, necessitating the heating of the ship, and which was done by means of steam. The evaporating apparatus consisted of two large rectangular boilers with several furnaces in each, for propelling the ship, and two auxiliary boilers of one furnace each; the latter discharging their products of combustion through the uptakes of the adjacent main boilers. One of the auxiliary boilers being out of use, steam was kept night and day on the other during the continuance of cold weather—about six months, and this boiler had previously been long used for distilling and other purposes, during all of which time, as well as during the winter in question, the products of combustion had been delivered into the uptake of the neighboring main boiler, and during most of the time no water had been in the latter.

At the end of the cold season, steam was raised in the main boilers for the purpose of enabling the vessel to resume cruising in other parts of the station, but pressure had hardly more than shown in them, before the uptake of that one through which the smoke and gases had long been passing, fell out, and the ship was obliged to proceed under the other main boiler, to a port where extensive repairs were made, and the iron of the damaged uptake was found to be extremely thin.

The next engineer-in-charge concluded that the rupture was due to the iron being "burnt" by comparatively low temperature continued through several months, almost as positively as if a crown sheet had been exposed for a few minutes to the furnace heat with no water on the other side.

In the next winter the vessel was again ordered to the same place, and the new engineer resolved to avoid a repetition of the trouble if his theory was correct. First, the disused auxiliary boiler was again put in operation, and employed alternately with the other for distilling water and auxiliary purposes, thus diminishing the heating of either main uptake by one-half. Next, a sufficient quantity of thin, cheap boiler iron was procured to form a partial guard in the uptake of either main boiler, and bent so as to cover the tube-sheet and the upper part of the smoke passage. The pieces were short enough to be passed

into and out of the uptake when required, the only fitting consisting in the slight overlapping of the edge of one sheet by the edge of the next, without any fastenings, until the partial channel reached as far as the smoke-pipe, each sheet resting on the lower edge of that part of itself which covered the tube-sheet, and being essentially an inverted U with one leg much longer than the other. This guard was put in position in a few minutes in either of the main uptakes, according as it was the turn of the auxiliary boiler on the same side of the ship to be used, and when there was no water in the corresponding main boiler.

It is believed that these precautions would have prevented a repetition of the weakening effects noted, and that similar means may be of use under certain similar conditions, for it is on record that steam was formed recently in a boiler containing water—but in the furnaces of which there were no fires—owing to the proximity of said boiler to others in which fires were maintained. But the matter was not put to the test, because in the following winter small stoves were used instead of steam heaters; and, by the way, were found more economical of coal, as all that would have been used for banked fires at night was saved, new fires being started in the morning.

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It was the intention to illustrate the engines of the *Charleston* in connection with the trial of that vessel, but, owing to circumstances of weather and to unforeseen difficulties in connection with the reproduction from photographic negatives, the work could not be completed in time for this issue of the JOURNAL.

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## ERRATA.

Table facing p. 236 of No. 3: In the column headed "When to be completed," *Petrel* should be Dec. 27, 1887, and *Baltimore* June 17, 1888. Limit of cost of Dynamite Cruiser No. 2 should be $450,000; of Cruisers Nos. 12 and 13 (now Gunboats 5 and 6), $350,000. Under the head "Type of boilers," for the *Puritan*, the data have been moved up one line, as have the succeeding data of the table for that vessel.